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THE DESIGN AND CONSTRUCTION OF A
HIGH VACUUM BEARING TEST APPARATUS

by
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Massachusetts Institute of Technology

B.S. in M.E.

May 21, 1962

Thesis
W22214

THE DESIGN AND CONSTRUCTION OF A
HIGH VACUUM BEARING TEST APPARATUS

by

MELVIN E. WALKER

Submitted in Partial Fulfillment

of the Requirements for the

Degree of Bachelor of Science

at the

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

June, 1962

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ACKNOWLEDGEMENTS

I would like to thank my thesis advisor, Prof. R. W Mann for his encouragement and guidance throughout this project.

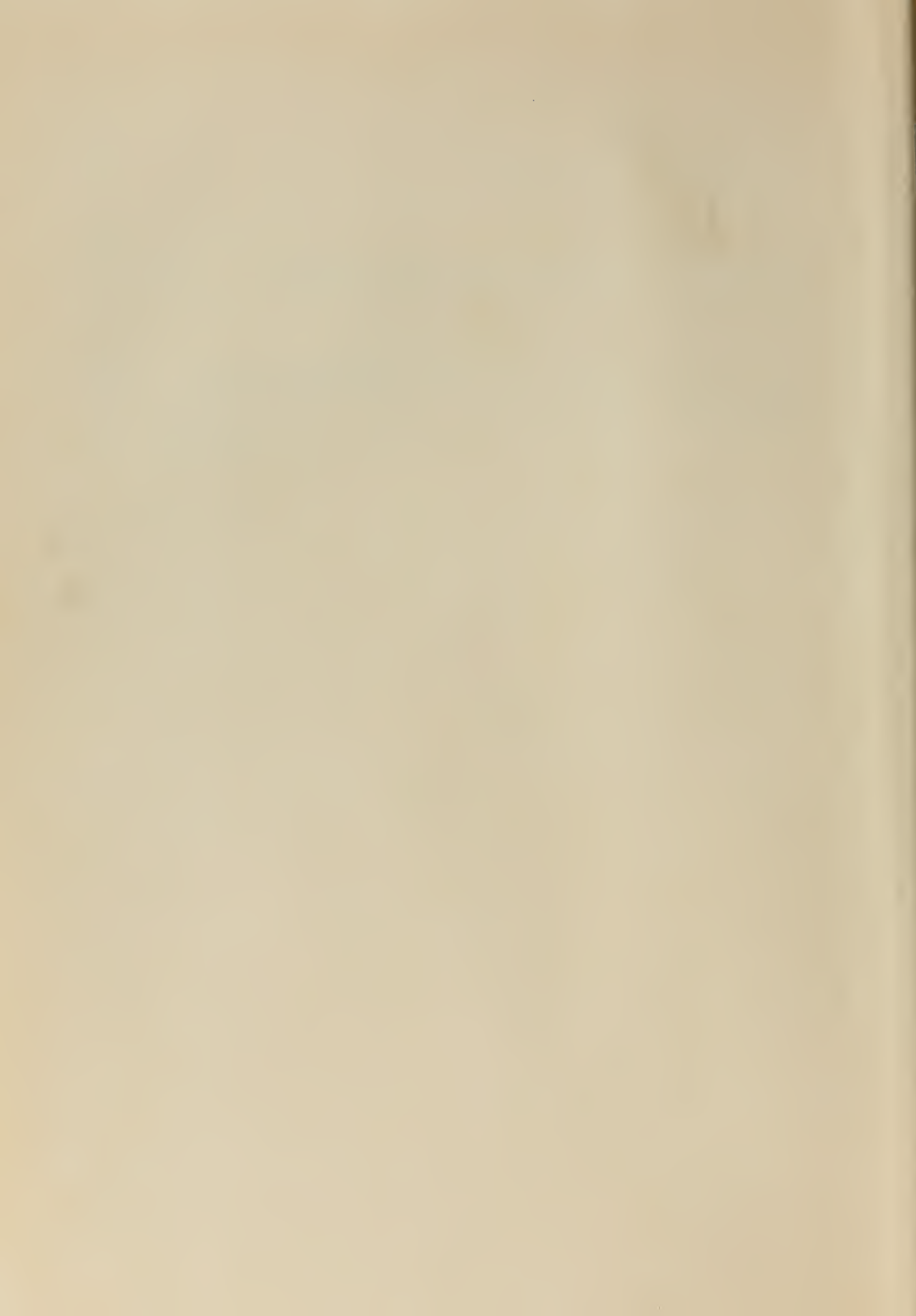
This thesis was sponsored by the Harvard Solar Satellite Project at the Harvard College Observatory. I would like to thank Nathan Hazen and the staff at the Harvard College Observatory for this opportunity and the consideration which they gave me.

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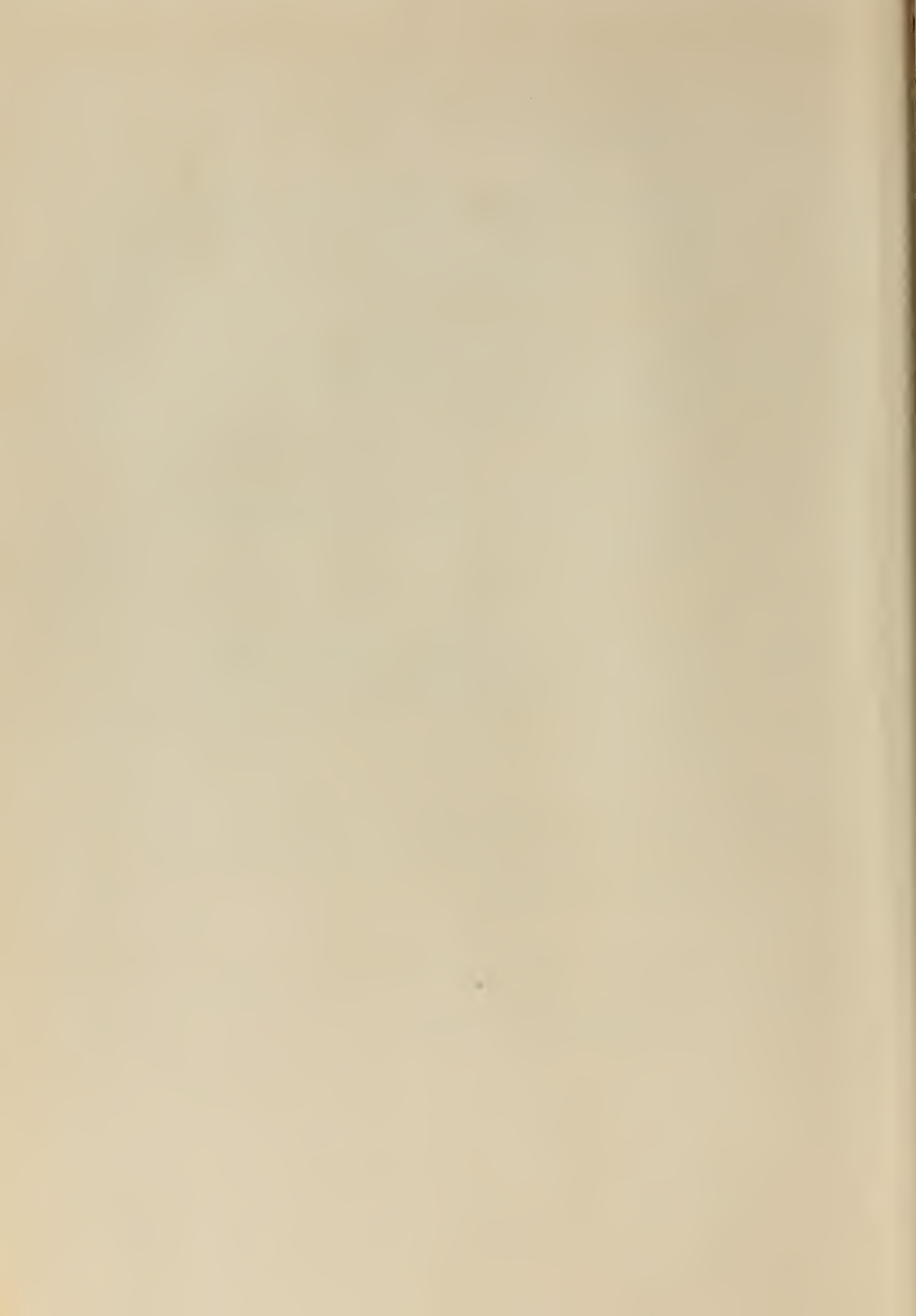


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ABSTRACT

The growing interest in the exploration of the universe makes mandatory new machines which must operate under the extreme temperatures and the high vacuum (10^{-6} to 10^{-13} mm Hg.) of outer space. A critical problem is that of developing bearings which will operate for long, unattended periods of time. Conventional bearings cannot be used because the liquid lubricants are quickly evaporated by the low pressure, and the solid lubricants wear or chip away, exposing the clean metal. If two perfectly clean surfaces come in contact, they weld together and then leave a rough surface when they are torn apart. This does not happen in our atmosphere because the metal surface quickly adsorbs films of oxygen and water vapor which act as lubricants.

There are many proposed designs for bearings which can operate under this high vacuum, but there is very little empirical data on the performance and useful life of these bearings. The purpose of this thesis is to design and construct an apparatus which will measure the performance of instrument bearings operating in a high vacuum.



1.1 INTRODUCTION

Man's ideas and ambitions have always been far in advance of the materials necessary to fulfill them. His greatest ambition today is to leave the earth and travel throughout the universe. In any vehicle traveling through space, friction, wear, and lubrication may well constitute the greatest materials problem.

1.2 BACKGROUND

The first extensive study of friction conditions in a vacuum was performed by T. P. Hughes in 1939. He found that the coefficient of friction of copper on copper in a vacuum of 10^{-6} mm Hg. was approximately 10 times the coefficient in air. This same experiment and similar experiments have been performed numerous times since 1939, but all these people were interested specifically in the mechanism of friction and wear.

There had been very little interest in bearing operation in a vacuum until the last few years. This interest was spurred by the development of the orbital satellite. I was first introduced to the problems of bearings operating in a vacuum at the Harvard College Observatory.

During my Senior year at M.I.T., I worked part time on the Harvard Solar Satellite Project. My work there was concerned primarily with the design of a grating drive system for an orbiting solar spectrometer. It was impossible to hermetically seal the spectrometer because it operated in a high ultraviolet frequency range which would not pass through a lens. This made it necessary for all the parts of the spectrometer to be exposed to the high vacuum. The need for bearings to operate in this grating drive system suggested the need for some means of testing bearings in a high vacuum. I chose this to be my Bachelor thesis problem.

1.3 THESIS PROBLEM STATEMENT

The purpose of my thesis is to devise a means for collecting empirical data on bearing performance under space-like conditions. The thesis consists of the designing and the construction of an apparatus that will continuously measure and record the running torque of bearings inside an evacuated bell jar.

2.1 CONVENTIONAL BEARING TECHNIQUES

Lubrication techniques are usually divided into two

general catagories, hydrodynamic lubrication and boundary lubrication. Hydrodynamic lubrication depends upon having a continuous fluid layer separate the two surfaces and exert sufficient pressure to keep them from touching. It is difficult to insure hydrodynamic lubrication on earth,, and even more difficult in space where the low pressure (10^{-6} to 10^{-13} mm Hg.) will make retention of the fluid a problem. In space there is also the increased difficulty of resupply of the lubricant because of long-time unattended operation and the absence of gravity. Moreover, there is the problem of lubricant depositing upon the surface of optical and electrical equipment that might be near by.

In boundary lubrication, the solid surfaces actually rub against each other. A liquid lubricant may or may not be present. On earth, exposed solid surfaces quickly adsorb films of oxygen and water vapor. The surfaces are thus "dirty", and the "dirt" acts as a lubricant and plays a major part in reducing friction under conditions of boundary lubrication. When metal surfaces are clean and free from adsorbed films of atmospheric gases, they commonly tend to weld to each other on contact; wear rates and coefficients



of friction are then very high. The cleaning of surfaces in space takes place by loss of adsorbed gases into the vacuum. Mechanical wear between mating surfaces provides a mechanism for removing any hard oxide coating that might have formed on the surfaces. Once the surfaces are clean, they will tend to stay clean in space.

It has been shown that both conventional types of lubrication are inadequate for long-term space application, but in space as on earth, the load and speed at which the bearing is operated greatly affect its useful life. There are a number of suggested bearing designs that appear to get around the space-imposed problems, but there are little empirical data that can be used in the designing of space vehicles.

2.2.1 PERFORMANCE DATA OF CONVENTIONAL BEARINGS

Every bearing catalog contains charts or nomagrams which aid in the selection of the proper bearing and lubricant for a given application. These charts give the approximate frictional torque and the expected life of the bearing for any combination of load and speed.

There is no analytical method known by which the torque and useful life of a ball bearing can be calculated. All

the performance data found in bearing catalogs were obtained by testing many bearings under a given operating condition and then performing a statistical analysis upon the data. It is necessary to do this at enough load-speed combinations so that smooth curves can be drawn between the points on a graph. These charts, therefore, apply only to well lubricated bearings operating in the earth's atmosphere. Thus, this data is not applicable in choosing a bearing intended to operate in outer space.

2.2.2 PERFORMANCE DATA OF BEARINGS IN A VACUUM.

There are many proposed solutions to the problem of unsatisfactory bearing operation in a high vacuum, but none of these have warranted the rigorous testing necessary to obtain a complete array of performance data.

To choose a bearing for a particular application, it is necessary to test several types of bearings under the actual operating conditions, and then choose the one that gives the most desirable combination of torque and useful life.

The purpose of the bearing test apparatus is to fulfill this need and to eventually collect enough data from

which the above mentioned charts can be completed for any bearing that looks promising for space application.

3. DESIGN CRITERIA

3.1 NUMBER OF TEST STATIONS

Since all bearing performance specifications are statistical, the more bearings tested, the more reliable the processed data will be. In the design of the test apparatus, space was of primary importance because the entire apparatus had to fit inside of a 14-inch bell jar. This limited the number of bearings that could be tested at one time. Another factor which must be considered is the complexity of the apparatus. If it is too complex, it is more subject to mechanical or electrical failure, and it is more difficult to set up for a test run. The complexity of the apparatus also increases its cost.

After considering all the above factors, I chose a criterion of 10 to 20 test stations.

3.2 TEST BEARING LOAD AND SPEED

Conventional instrument bearings are tested at radial and axial loads of 0 to 30 lb, and speeds up to 30,000 rpm.

This results in an overall frictional torque range of 100 to 100,000 milligram millimeters.

I expect bearings in a vacuum to be operated at much lighter loads because of the effect of elastic deformation of the bearing material. The deformation causes the hard oxide coating of the metal to crack and thus exposes the clean metal. For this reason, I thought a maximum radial load of 10 lb and an axial load of 5 lb would be adequate for the testing of instrument bearings in a vacuum.

I chose a maximum test speed of 2000 rpm. because this includes the speed of most electric motors and because I felt a higher speed would introduce vibration caused by unbalanced rotating masses.

3.3 MEASURABLE TORQUE RANGE

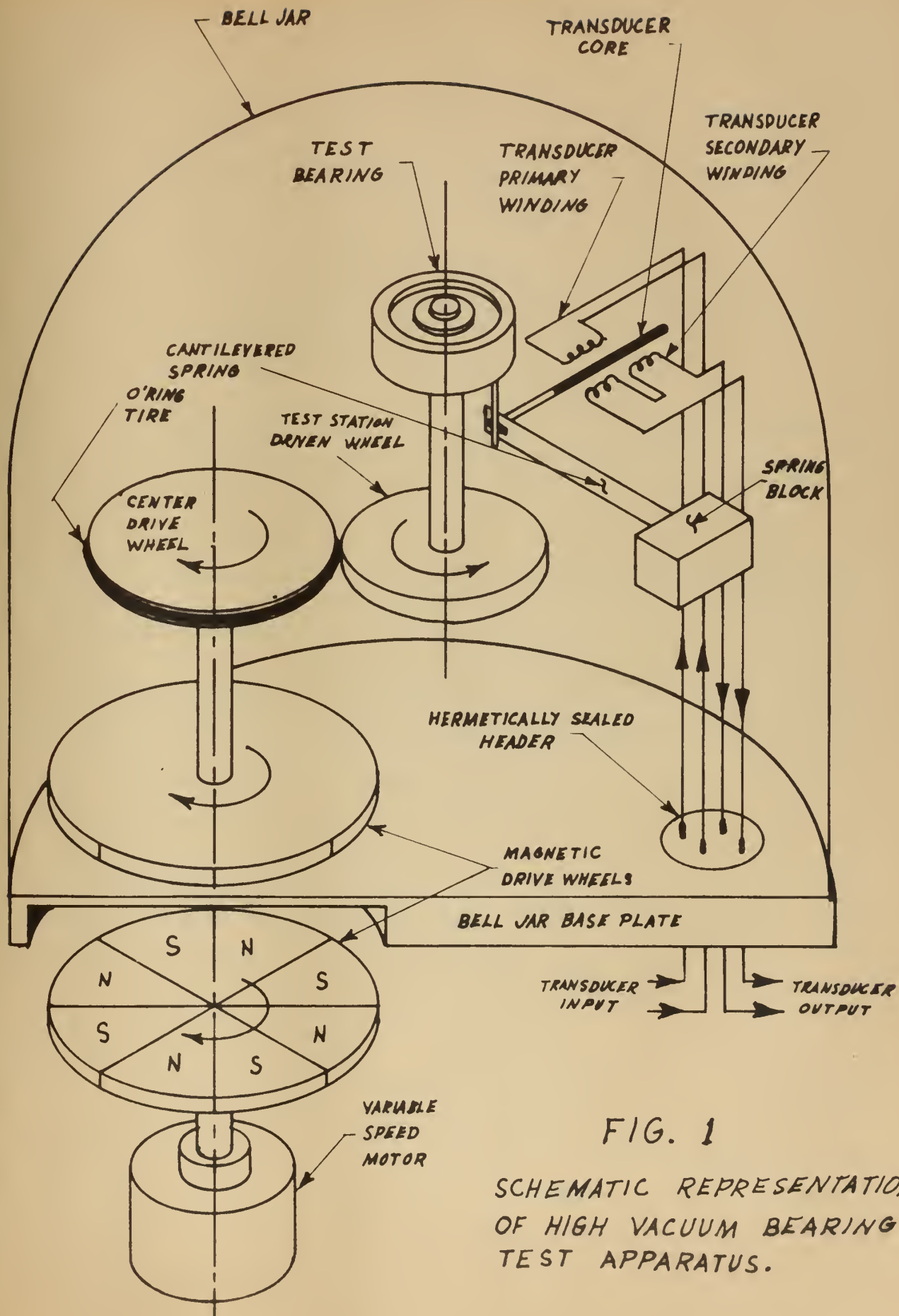
Although I am testing at lighter loads and slower speeds than those listed in the catalog, I expect higher torques of bearings operating in a vacuum. This will be especially true if journal bearings are used. For this reason I think the apparatus should be capable of measuring torques from 100 to 1,000,000 milligram-millimeters.

3.4 TEST PRESSURE

Ideally, I would like to test the bearings at a pressure of 10^{-13} mm Hg. which is the pressure beyond 4000 miles from earth. A pressure this low cannot be reached in the laboratory without special equipment and conditions which are not compatible with the tests I wish to perform. Therefore, I have to be satisfied with a pressure of 10^{-7} to 10^{-8} mm Hg. which I can obtain. This pressure corresponds to approximately 250 miles from earth which is equal to or greater than the orbital radius of most of our satellites.

4.1 FUNCTIONAL DESCRIPTION OF BEARING TEST APPARATUS

The drive motor which supplies the torque to the test bearings is mounted under the bell jar base plate. (see Fig. 1) A magnetic drive transmits rotational motion through the base plate to the center drive wheel. A small driven wheel on each test station rides against the center drive wheel. The inner race of the test bearing is rotated by the shaft which passes through the test station driven wheel. A cantilevered spring is used to keep the outer race from rotating. The displacement of the spring is proportional



to the bearing frictional torque, and the displacement of the transducer core attached to the spring determines the amplitude of the signal out of the transducer. Therefore, the signal out of the transducer is proportional to the bearing frictional torque and it can be displayed on a meter or recorded on a strip chart recorder.

The apparatus will be used for the selection of the best type of bearing for a particular application. This will be accomplished by setting the speed and load equal to that of the actual situation and inserting the different types of bearings into the eight test stations. By running tests at a number of combinations of load and speed, bearing torque (or bearing life) vs. rotational speed can be plotted with bearing load as a third parameter.

4.2 COMPONENT DESCRIPTION

The bearing test apparatus is made up of a drive system, eight bearing test stations, and a base plate.

4.2.1 THE DRIVE SYSTEM

The drive system consists of a variable speed DC motor, a magnetic coupling, and a central drive wheel (see Fig. 1). The DC motor (90 to 3000 rpm) is mounted vertically underneath

the bell jar base plate which is a one-inch thick aluminum plate. A recess is milled in the bottom of the base plate to permit the two wheels of the magnetic coupling to be mounted $7/32$ inches apart. In the set-up shown in Fig. 2, a plain sheet of $3/16$ aluminum plate was used to test the drive system.

Two identical magnet wheels are used to transmit the torque through the base plate (see Fig. 3). One is mounted on the motor shaft and the other one is mounted on a shaft running vertically through the test apparatus base plate. The magnet wheel consists of an aluminum disc with 65 horseshoe magnets mounted in an array such that 13 alternating north and south poles are formed around the disc. It is important to always have like poles adjacent to each other on the disc or some of the magnetic flux lines will be between adjacent magnets rather than through the base plate to the opposite wheel. With a spacing of $7/32$ in. between the wheels, a torque of 40 in. oz. (2880 gm cm) can be transmitted by the magnetic coupling.

The center drive wheel which provides the rotational motion to each of the eight test stations is locked to the

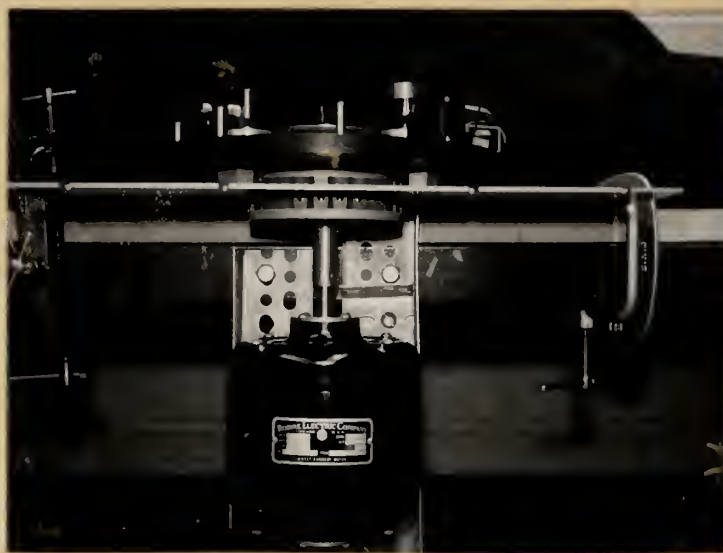


Fig. 2 Set-up by which the apparatus drive system
was tested.



Fig. 3 One of the magnet wheels of the magnetic
coupling.

upper end of the shaft holding the driven magnet wheel. A buna-N o'ring is used as a tire on the center drive wheel to prevent metal to metal contact with the test station wheel. It also prevents vibration from being transferred to the test stations.

4.2.2 THE TEST STATION

The test station block supports a vertical shaft which has locked to it, the wheel that is driven by the center drive wheel. (see Fig. 4 and Dwg. D-5101) The test bearing is placed in a housing and slipped onto the top end of this shaft; the inner race rotates with the shaft, but the outer race is held stationary by a pin in the housing coming in contact with a cantilevered spring. The greater the frictional torque, the more the end of the cantilever spring is displaced. This displacement is measured by a variable reluctance transformer and can be displayed on a meter or recorded by a strip-chart recorder.

To calibrate the bearing test station, it is placed on the calibration jig as shown in fig. 5. The shaft of the test station is brought in contact with a flat on the



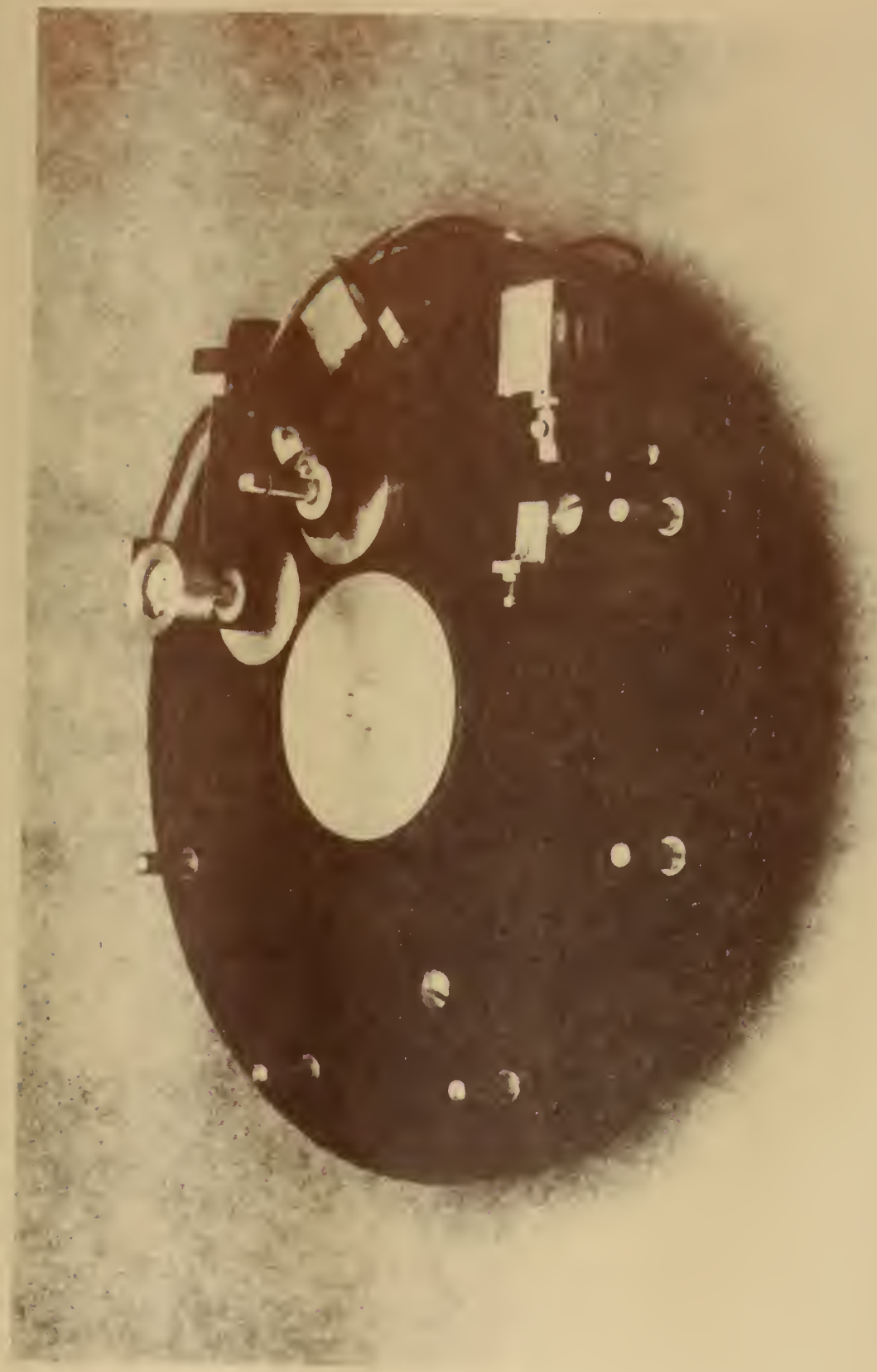


Fig. 4 Bearing test apparatus showing the base plate, the center drive wheel and two test stations.



Fig. 5 Test station calibration jig being used to locate knife-edge.

calibration jig. To insure that the force from the bearing housing pin is perpendicular to the cantilevered spring, the knife-edge of the test station is placed on the centerline of the test station block. This is accomplished by adjusting the spring block until the test station knife-edge comes in contact with the one on the calibration jig. Accurate adjustment is obtained by placing an 0-80 screw through the hole of the adjustment bracket of the calibration jig and screwing it into the spring block (see Dwg. B-5105). When the spring block locking screw is loosened, the block can be accurately moved back and forth by turning the 0-80 screw.

To calibrate the transducer of the test station, the calibration jig is placed on its side as shown in fig. 6. The input to the transducer primary is adjusted to a prescribed frequency and amplitude. Weights are then hung on the knife-edge of the test station and the transducer output voltage is read. A force vs. voltage curve can be drawn. The force can be converted to torque by multiplying it by $1/4$ in., the torque arm of the bearing housing pin. In

fig. 6, the transducer output voltage was measured on an oscilloscope because an output demodulator had not yet been obtained.

After the transducer has been calibrated, the calibration jig is turned upright again. The test bearing and housing can now be put in place. The bearing is subjected to a radial load by a magnet as of yet not received from the factory. The magnet bracket is held in place by two screws in the holes next to the bearing spring block. The magnetic pull on the bearing housing will be adjusted by changing the separation between the magnet and the bearing housing (see Dwg. D-5101). A record of force vs. separation will have to be made and this will then be used to apply the proper radial load to the bearing. The test station is now ready to be placed on the base plate.

The three calibrations discussed above need be performed only once. The bearing housing can be lifted off without disturbing the magnet, the transducer, or the spring block.



Fig. 6 Calibration jig being used to calibrate torque measuring transducer.

4.2.3 THE BASE PLATE

The base plate supports the center drive wheel which supplies motion to the eight test stations. Eight vertical dowl pins are provided as supports for the test stations (see Fig. 4). A spring plunger is used to hold the test station in contact with the center drive wheel. If one test bearing has failed, that test station can be uncoupled from the center drive wheel by energizing a solenoid. This pivots the test station on its dowl pin support and pulls it away from the center drive wheel. When the test station

is pulled away, a pin drops over the edge of the plunger bracket and holds the test station in place. The solenoid can then be de-energized.

5. OPERATION PARAMETERS OF THE BEARING TEST APPARATUS

After the design was completed, the following operating parameters were available.

Bearing Test Stations	8
Bearing Test Speed	90 to 2000 rpm
Radial Load	0 to 5 lb
Axial Load	.1 to 1 lb
Measurable Torque	100 to 300,000 mg-mm
Test Pressure	10^{-7} to 10^{-8} mm Hg.

Eight bearing test stations was the maximum number I could accomodate without making it necessary to recalibrate each torque measuring device before each test run.

The radial and axial loads must be applied without imposing any torque upon the test bearing. This can be accomplished only by the application of a body force on a symmetrical body. An axial load can be applied by placing weights upon the bearing housing. The bulkiness of the

weights limit the axial load to approximately one pound.

A magnetic force is the only practical body force that can be used to apply a radial load to the test bearing. This limits the radial load to approximately 5 pounds.

The transducer calibration yielded a measurable torque range of 100 to 300,000 mg-mm. If it is found desirable to increase this torque range, the spring specifications can be calculated as prescribed in the appendix.

6. DISCUSSION OF EXPERIMENTAL WORK

6.1 DRIVE SYSTEM

The drive system has been tested and found to transmit 40 in. oz (2880 gm cm) of torque. This is approximately three times the calculated torque required.

Originally there was a 1/8 in. o'ring on the center drive wheel and the test station wheel was pushed against it with constant force. This caused the test station to follow every irregularity in the tire. By replacing the o'ring with a thicker one (3/16 in. wide) and by placing an adjustable stop on the inside of the test station, the

vibrations were eliminated. The thicker o'ring permitted, more deflection of the rubber. The stop held the test station wheel from following every depression in the tire, yet the tire would expand enough so that the two wheels never broke contact.

6.2 TRANSDUCER CALIBRATION

The transducer was first calibrated by placing a micrometer head against the test station knife-edge. The displacement sensitivity was found to be 0.05 volts/0.001 in.

A force calibration was made by hanging weights upon the test station knife-edge. This produced a force sensitivity of 0.040 volts/gram, or a torque sensitivity of .063 volts/gm cm. In the English system, this is 4.54 volts/in. oz. All of the calibrations made were linear within the accuracy of the oscilloscope voltage readings.

6.3 BEARING TESTING

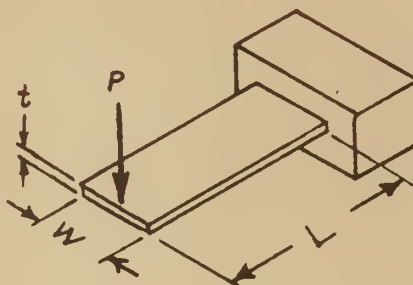
I have two completed test stations. As soon as the vacuum system is put back together, I plan to test two dry ball bearings. If the data looks reliable, I will have the

other 6 test stations manufactured and proceed on a test schedule for testing the performance of solid lubricants in a vacuum.

APPENDIX I. TRANSDUCER SPRING CALCULATION

The range of the measurable torque of the bearing test station can be adjusted to cover most any range simply by changing the spring dimensions. The displacement of the end of the spring (y) is small compared to the spring length(L), therefore a classical cantilevered beam equation can be used.

$$y = \frac{4 \times P \times L^3}{E \times W \times t^3}$$



Choose the maximum displacement equal to 0.030 in..
at maximum torque to be measured.

$P = \text{max. torque (in. lb)}/0.250 \text{ in.}$

$y = 0.030 \text{ in.}$

$L = 1.875 \text{ in.}$

$E = \text{Youngs Modulus of spring material.}$

$W = \text{Spring width (in.)}$

$t = \text{Spring Thickness (in.)}$

All the parameters of the above equation are fixed except for W and t . Any combination of W and t can be used so long as W does not exceed 0.250 in.

APPENDIX II. CHOICE OF DRIVE SYSTEM

There are many reasons for choosing this particular drive system. First, since any motor must have some type of bearings, it is questionable as to how long a motor could successfully operate in a vacuum. Second, there is no air surrounding the motor in a vacuum, and therefore, there is no means of cooling other than radiation. Third, this source of heat inside the bell jar is undesirable because it makes it more difficult to reach low pressure. Fourth, it is impractical to have a vacuum tight mechanical feed-thru at these low pressures. This leaves very little choice other than a magnetic coupling. The magnetic coupling has the further advantage that it transmits no mechanical vibration from the motor to the test apparatus.

APPENDIX III. CHOICE OF TORQUE MEASURING DEVICE.

There are numerous direct reading torque meters on the market, but these are quite expensive and are not designed to operate at extremely low pressures. Strain gage type transducers were disregarded because one gage would not cover the required range without going to very expensive specials. Variable capacitance type gages were considered, but these were found also to be very expensive and rather short lived with capacitance changing with time. For these reasons, the L.V.D.T. which is a standard item except for the teflon coated leads, was chosen.

The transducer is a linear variable-reluctance differential transformer, Sanborn model #595 DT-50. It is sensitive to 1 microinch of core displacement, but in my case a resolution of only 100 microinches is required.

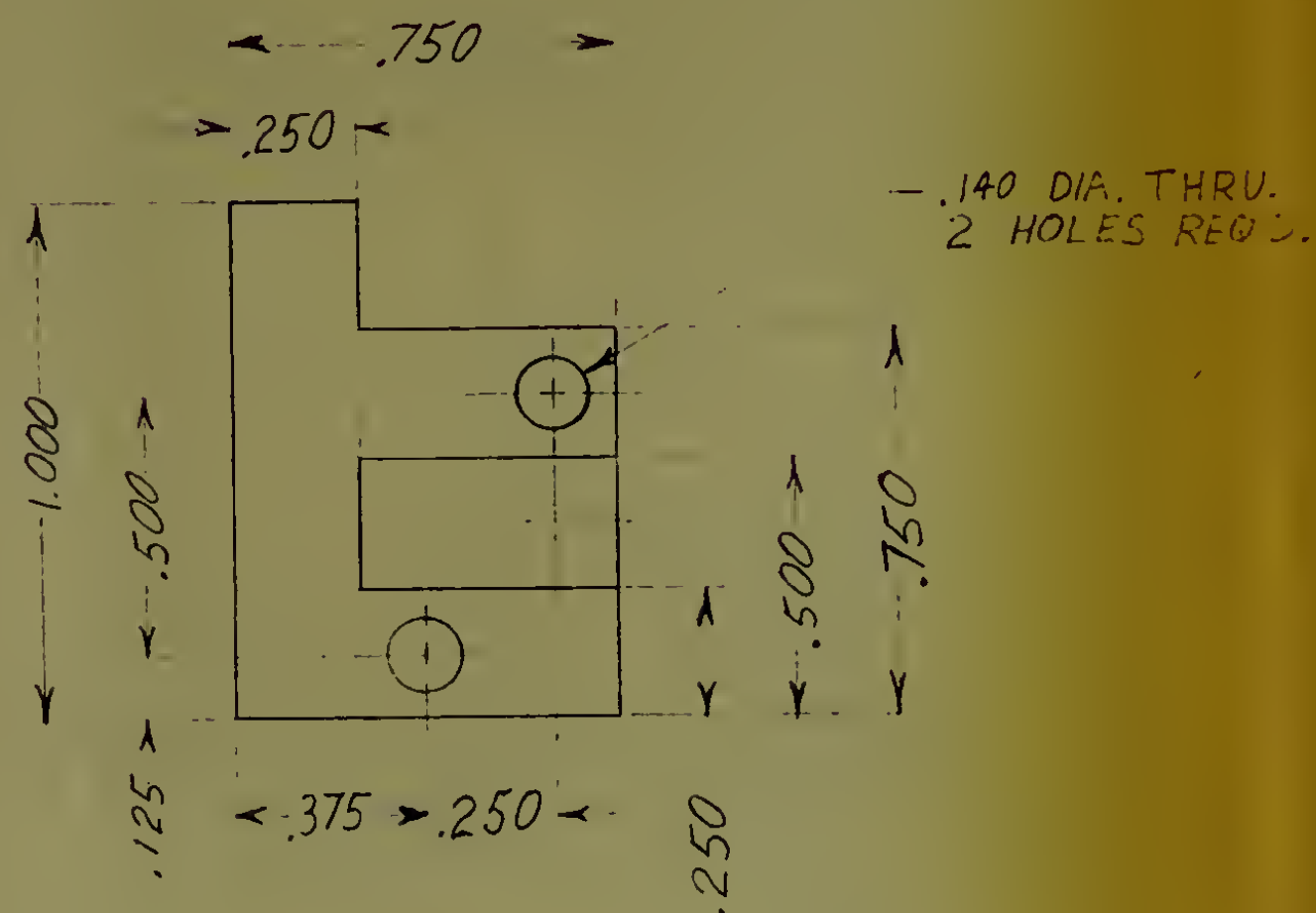
Another reason for choosing this transducer is that the core can be cantilevered from the spring and thus never come in contact with the transducer coil. This eliminates all friction in the torque measuring device except for the housing pin rubbing against the knife-edge. Most torque

meters depend upon having bearing with a frictional torque an order of magnitude less than the smallest torque measured.

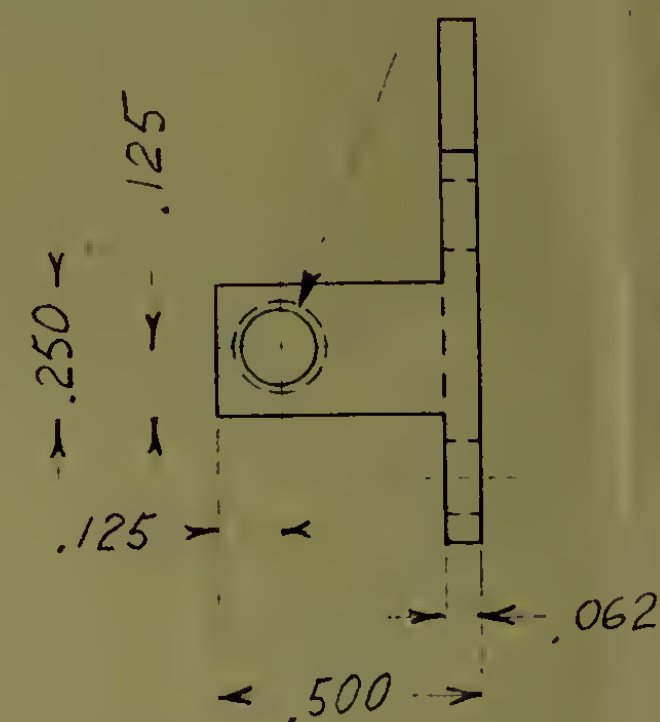
APPENDIX IV CHOICE OF TEFLON BEARINGS FOR APPARATUS.

Teflon bearings were chosen to be used in this bearing test apparatus because teflon is rather stable in a high vacuum, and it has low frictional torque for a dry journal bearing. The torque is still relatively high when it is compared with the torque of ball bearings; this is the reason it is not applicable to many situations where the power consumed is of prime importance. Teflon is difficult to machine to close tolerances; this also makes it undesirable because it is difficult to get the proper fit between the bearing and the shaft.

The teflon bearings, used in this test apparatus, were made to standard bearing dimensions; if a suitable ball bearing can be found for high vacuum operation, it will be a simple matter to change the bearings.



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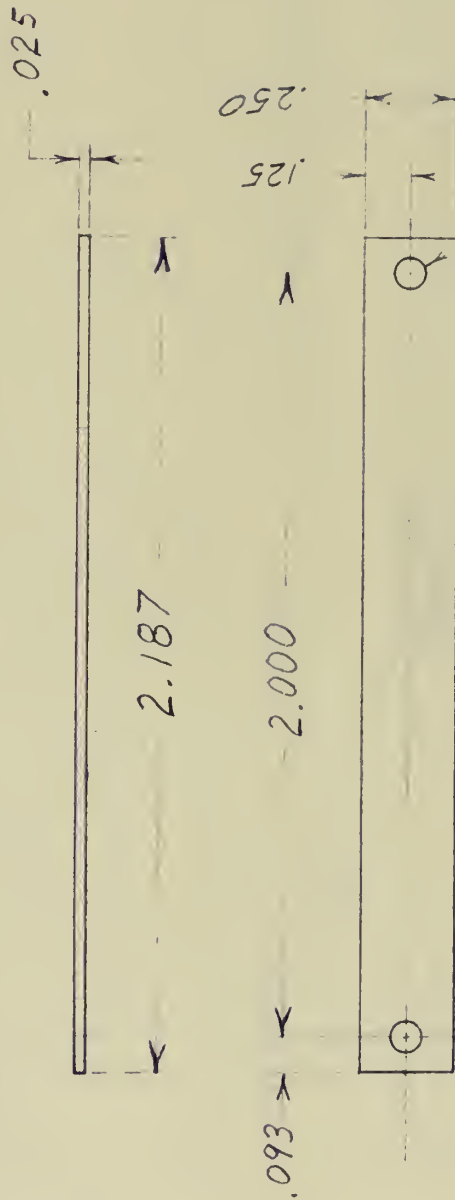


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FRACTIONS	DECIMALS	APPLICATION		 HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS		
$\pm \frac{1}{64}$	$\pm .005$	NEXT ASSEMBLY	USED ON			
MATERIAL & SIZE		D-5101			TITLE	
2024-T4 ALUMINUM		APPR	NLH	5/17/62	BRACKET, PLUNGER	
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MATERIAL & SIZE 1095 SPRING STEEL		APPR NCH 5/17/62		TITLE SPRING	
FINISH NONE		CHECK		SCALE 2/1	
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$.025 \pm .002 \times .250$ DEEP

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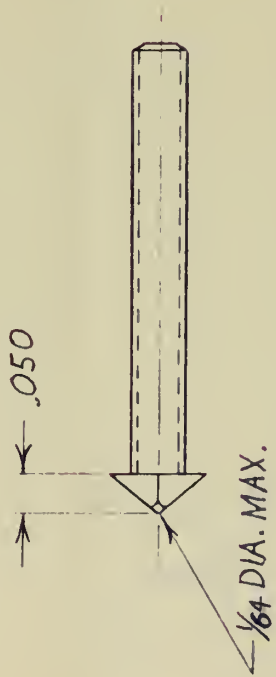
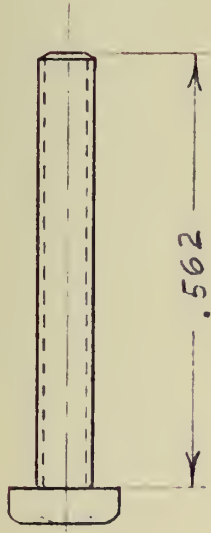
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BREAK ALL
FOUR CORNERS.

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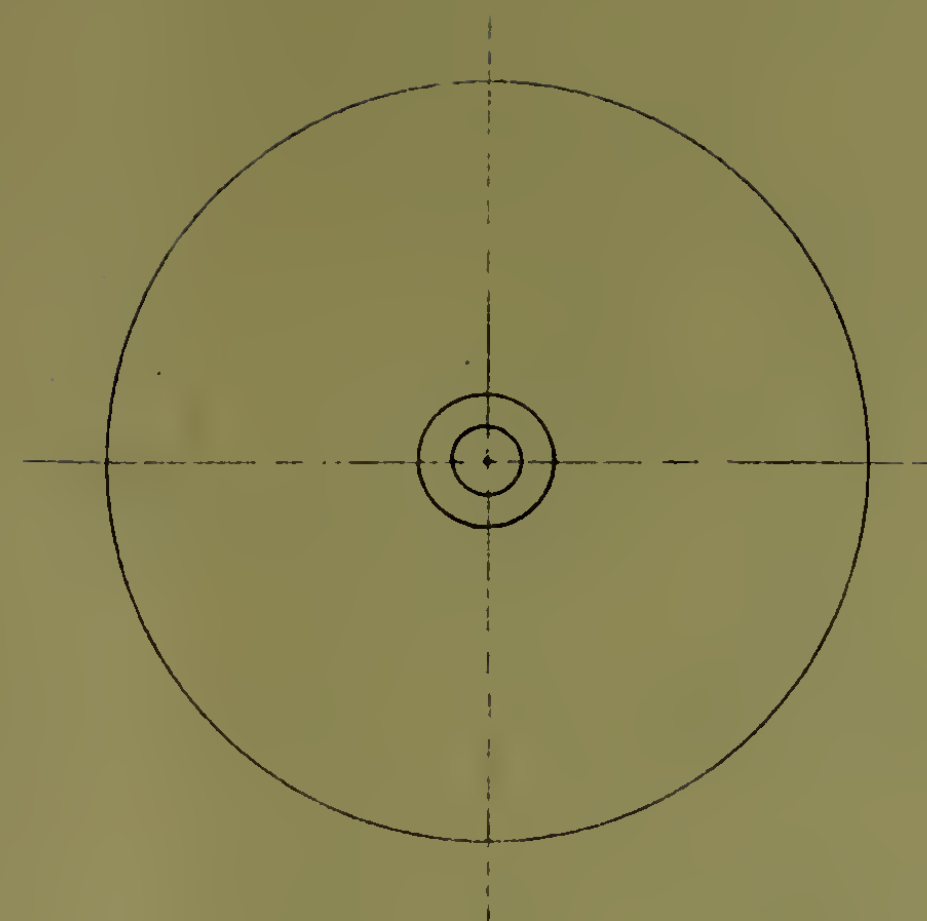
NOTE:
REMOVE ALL BURRS
& SHARP EDGES.

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MATERIAL & SIZE 2024-T4 ALUMINUM		APPR NLH	5/17/62	TITLE BLOCK, SPRING	
FINISH GOLD ANODIZED		CHECK		SCALE 2/1	REV LTR
		DRAWN <i>W. J. Walker</i>	12-10-61	NUMBER A-5203	



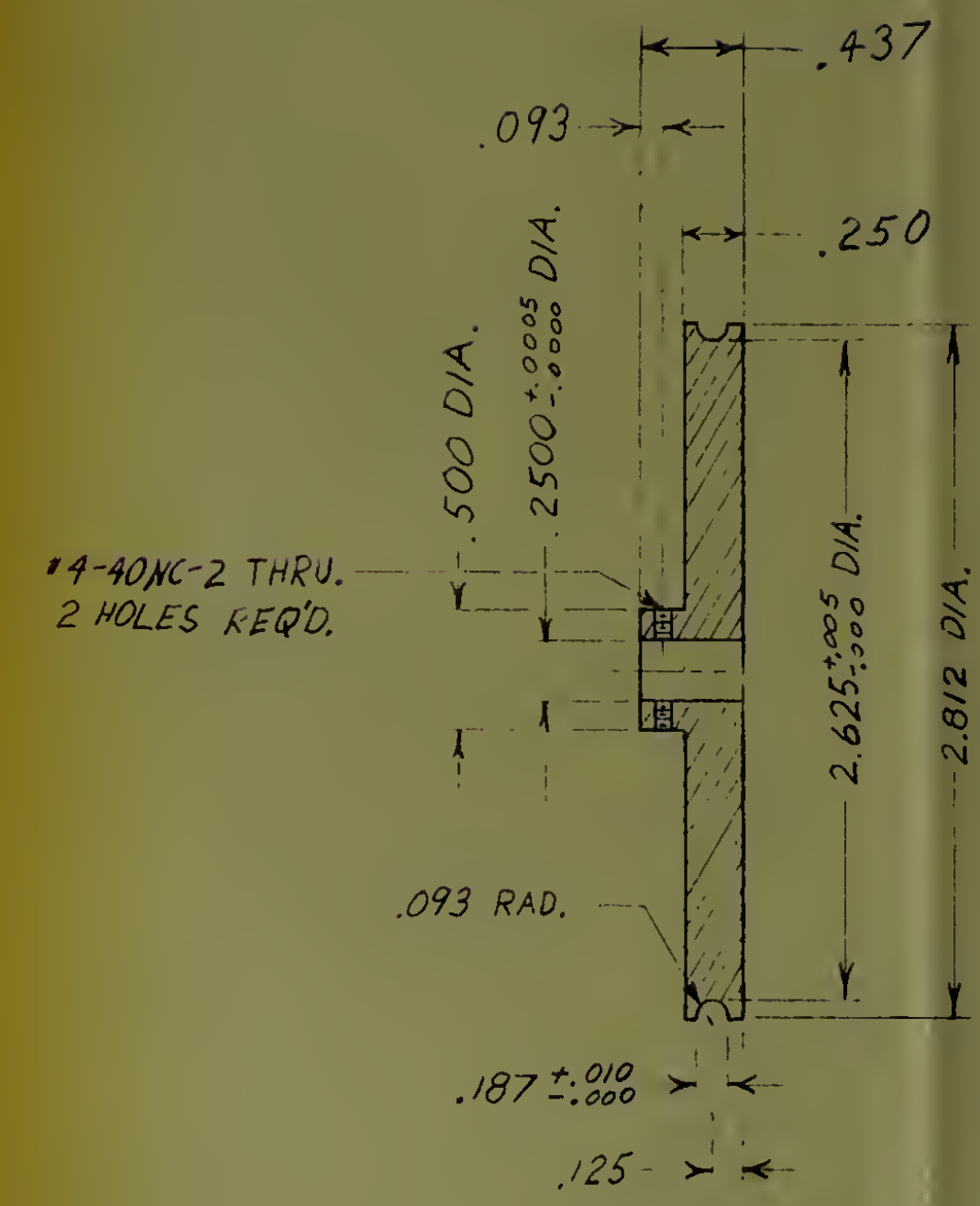
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MATERIAL & SIZE # 1-72NF-2 x 1/2 LG, HEX. HD. S.S.		APPR NLI-1	5/17/62	TITLE KNIFE EDGE	
		CHECK		SCALE 4/1	REV LTR
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REVISIONS		
REV	DESCRIPTION	DATE



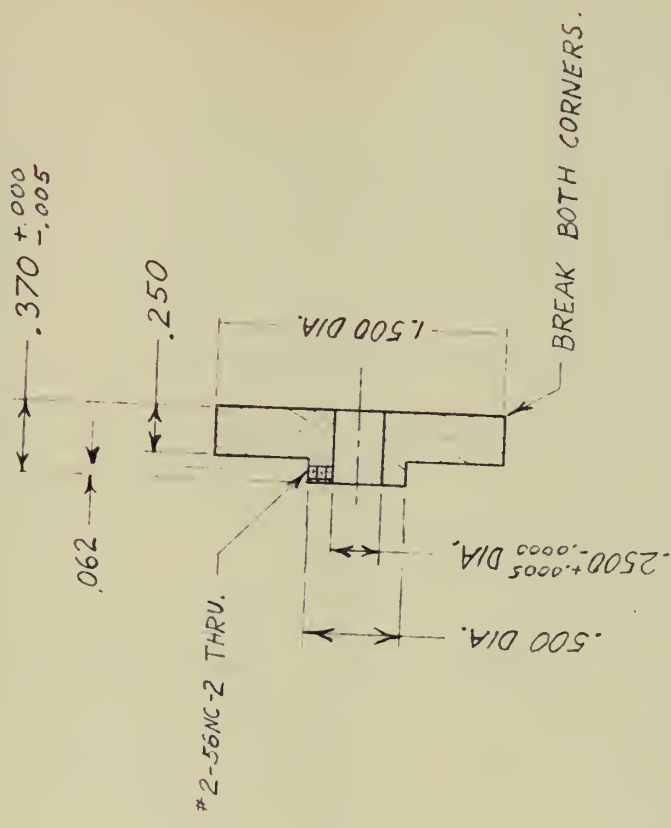
REV
NUMBER
B-5205

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm \frac{1}{16}$ $\pm .005$ $\pm \frac{1}{2}$		NO	QUAN	NUMBER	DESCRIPTION	
MATERIAL B SIZE 2024-T4 ALUMINUM		APPLICATION NEXT ASSEMBLY USED ON D-5101			 HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	TITLE WHEEL, DRIVING
FINISH CLEAR ANODIZED		APPR	NLH	5/17/62		
		CHECK				
		DRAWN	McWaters	3-16-62	SCALE 1/1	NUMBER B-5205
		SIGNATURE		DATE	REV LTR	

REVISIONS		
REV	DESCRIPTION	DATE

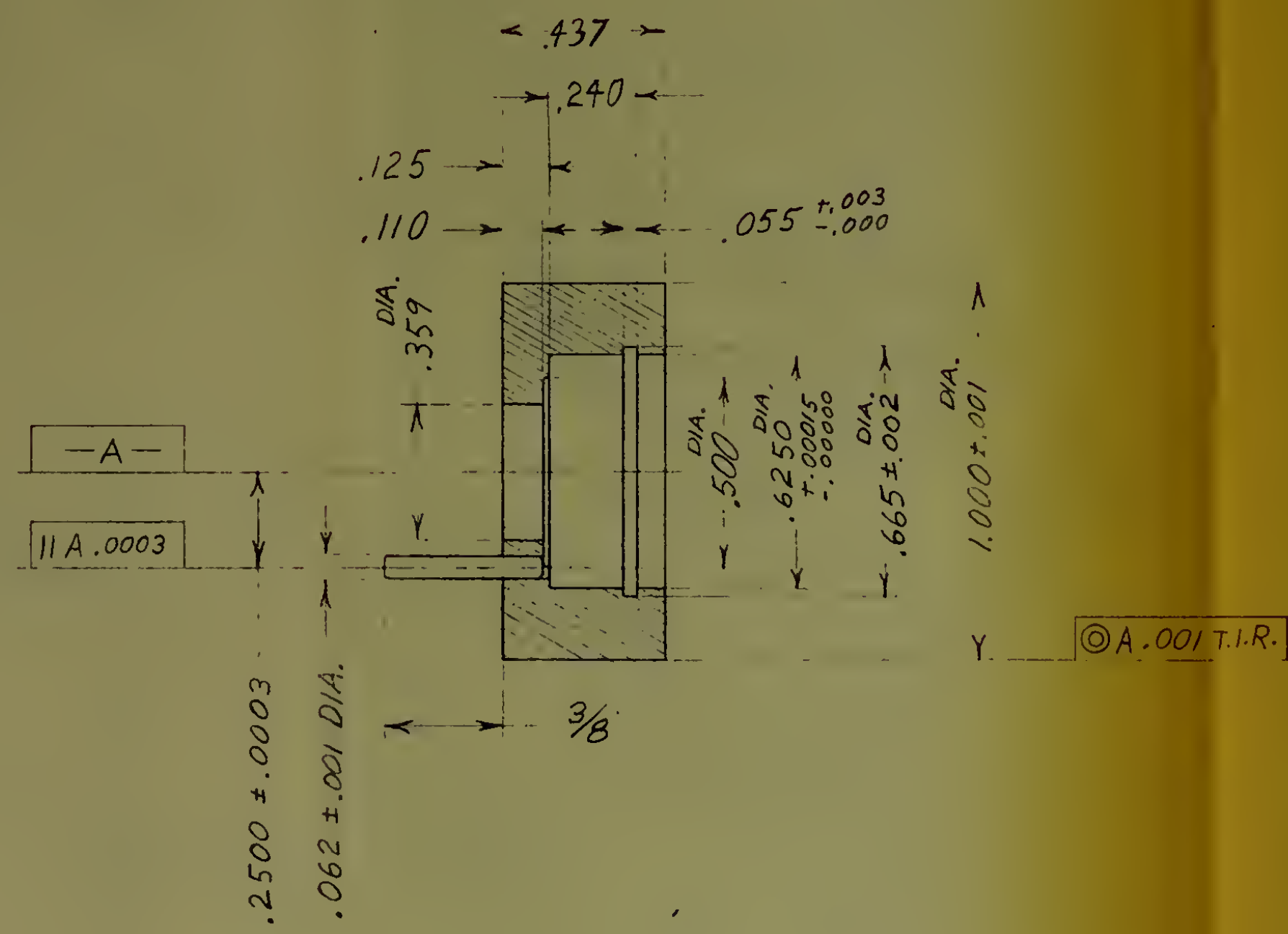


UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON		APPLICATION	
FRACTIONS	DECIMALS	ANGLES	USED ON
$\pm \frac{1}{16}$	$\pm .005$	$\pm \frac{1}{2}$	
MATERIAL & SIZE		NEXT ASSEMBLY	
2024 T4 ALUMINUM		D-5101	
FINISH		APPR	5/17/62
CLEAR ANODIZED		CHECK	
		DRAWN	3-16-62
		SIGNATURE	
		DATE	
		SCALE	1/1
		NUMBER	A-5206
		REV LTR	

HARVARD COLLEGE	
OBSERVATORY	
CAMBRIDGE 38 MASSACHUSETTS	
TITLE	
WHEEL, DRIVEN.	

REV
NUMBER
B-5207

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED

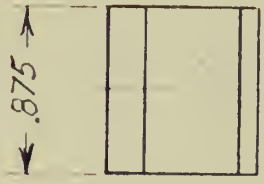
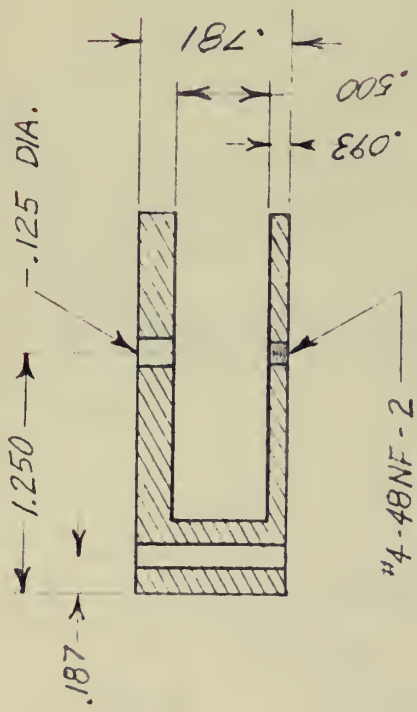
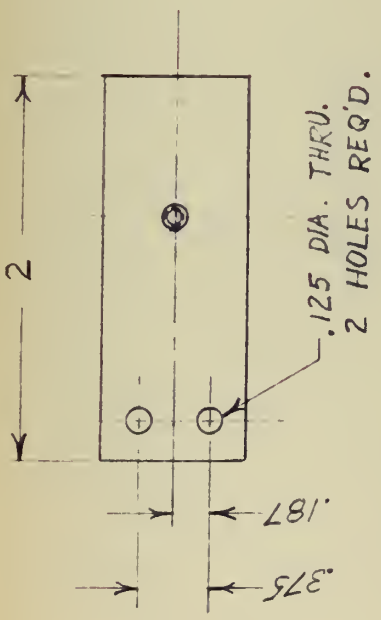


- NOTES:
1. REF. LINE -A- IS ϕ OF .6250 DIA.
 2. REMOVE ALL BURRS & SHARP EDGES.

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON		NO	QUAN	NUMBER	DESCRIPTION	
FRACTIONS	DECIMALS	ANGLES	APPLICATION			HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS
$\pm 1/64$	$\pm .005$	$\pm \#$	NEXT ASSEMBLY	USED ON		
MATERIAL & SIZE			D-5101		TITLE	
HOUSING:			APPR	NLM	5/17/62	HOUSING, TEST BEARING.
ANNEALED 1020 STEEL.			CHECK			
PIN: 303 STAINLESS.			DRAWN	McVinn	2-2-62	
FINISH			NONE		SIGNATURE	DATE
					2/1	
					NUMBER	REV LTR
					B-5207	

REVISIONS

REV	DESCRIPTION	DATE	APPROVED



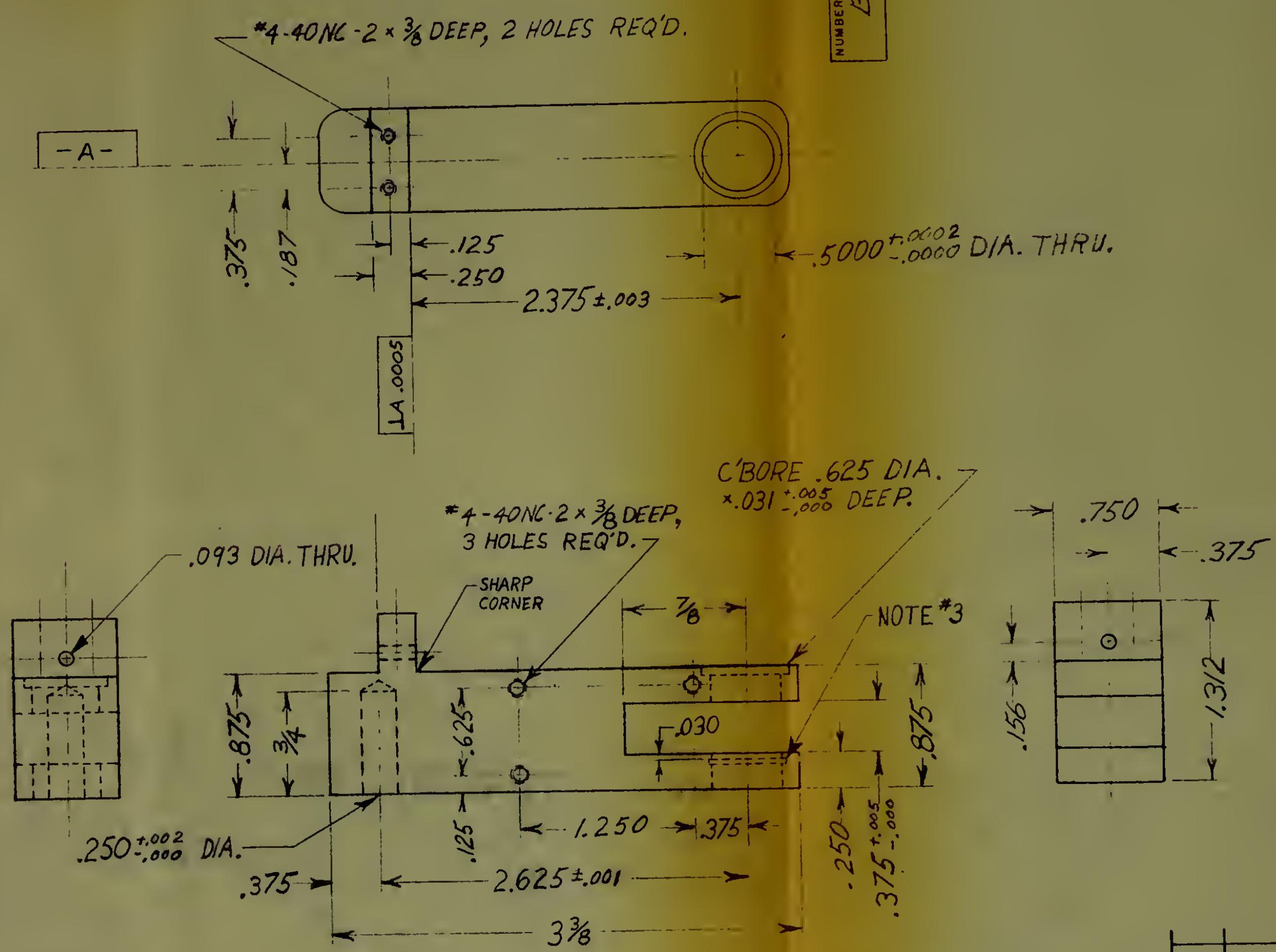
UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES	
$\pm \frac{1}{64}$	$\pm .005$
MATERIAL & SIZE	
2024-T4 ALUMINUM	
FINISH BLACK ANODIZED	

APPLICATION	
NEXT ASSEMBLY	USED ON
D-5101	
APPR	NLH
CHECK	
DRAWN	
DATE	
5-17-62	

HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
TITLE BRACKET, MAGNET	
SCALE 1/1	NUMBER B-5208
REV LTR	

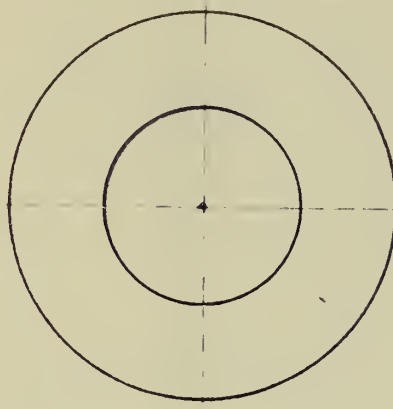
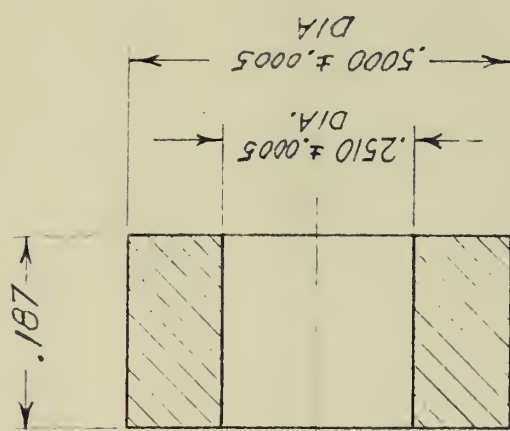
REV
NUMBER
B-5209

REVISIONS		DATE	APPROVED
REV	DESCRIPTION		



- NOTES:
1. REF. LINE -A- IS ϕ OF .250 & .500 BORE.
 2. REMOVE ALL BURRS & SHARP EDGES.
 3. SNAP RING GROVE DIMENSIONS. .530 $\pm$.002 DIA $\times$.039 $\pm$.002 WIDE.

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm 1/64$ $\pm .005$ $\pm \#$		NO		QUAN	NUMBER	DESCRIPTION	
MATERIAL & SIZE 2024-T4 ALUMINUM		APPLICATION		NEXT ASSEMBLY		USED ON	
		D-5101					
FINISH VAPOR BLAST & BLACK ANODIZE.		APPR	N/LH	5/17/62		TITLE	
		CHECK				BLOCK, BEARING TEST STATION	
		DRAWN	Mel Walker		1/29/62		SCALE
		SIGNATURE		DATE		NUMBER	
						B-5209	
						REV LTR	



REVISIONS		
REV	DESCRIPTION	DATE

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES		APPLICATION	
$\pm$ $\frac{1}{16}$	$\pm .005$	NEXT ASSEMBLY	USED ON
MATERIAL & SIZE		D-5101	
TEFLON		APPR	NLH
		CHECK	
FINISH		DRAWN	W. H. H. H.
NONE		SIGNATURE	
		DATE	2-28-62
		SCALE	4/1
		NUMBER	A-5210
		REV LTR	

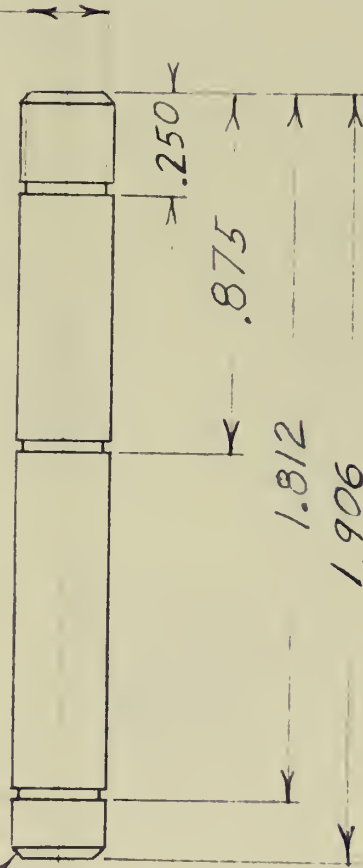
HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
TITLE BEARING, TEFLON, PLAIN.	



-CHAMFER .020x45°, BOTH ENDS.

GROOVE DIA., .220 ± .002 TYR.
 .002 T.I.R.
 .029 ± .003 TYR.
 .000

.24975^{+0.0000}_{-.0001} DIA.

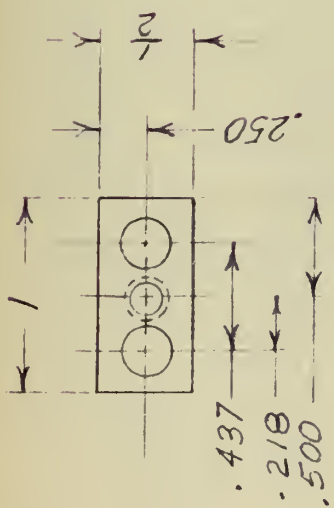


NOTE.
 REMOVE ALL BURRS &
 SHARP EDGES.

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES ± 1/64 ± .005 ± 3°		APPLICATION NEXT ASSEMBLY USED ON D-5101		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
MATERIAL & SIZE PIC ULTRA PRECISION SHAFTING # A6-20		APPR NLH CHECK DRAWN M. Whelan		TITLE SHAFT, TEST BEARING	
FINISH NONE		DATE 5/17/62		SCALE 2/1	
		SIGNATURE		NUMBER A-5211	
				REV LTR	

REVISIONS

REV	DESCRIPTION	DATE	APPROVED



C'BORE .250 DIA
x 3/8 DEEP
2 HOLES REQ'D.



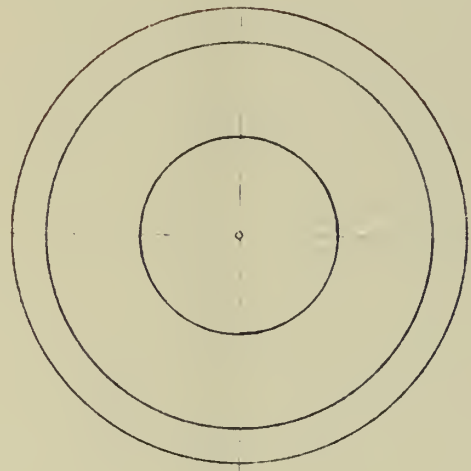
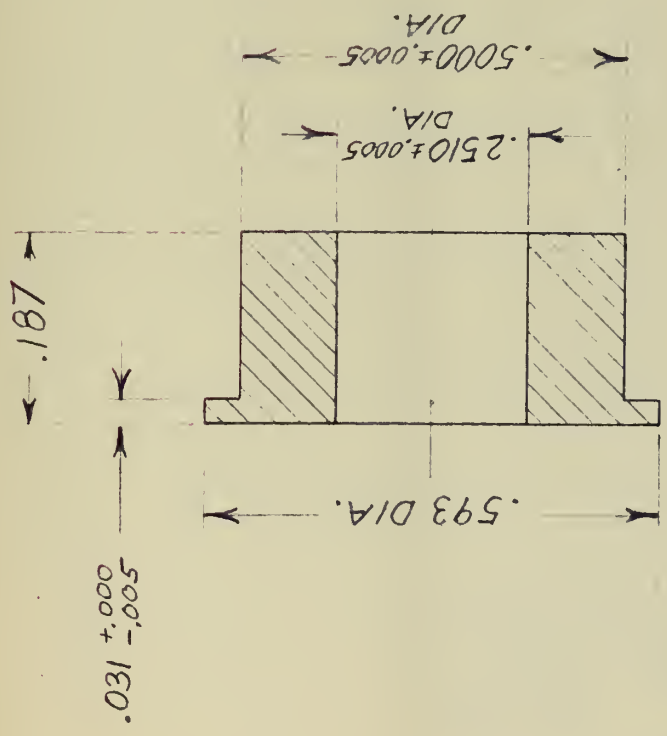
5/32 DIA. THRU.
2 HOLES REQ'D.

#1/4-20NC-2 THRU



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm 1/64$ $\pm .005$ $\pm \#$		APPLICATION NEXT ASSEMBLY USED ON D-5101		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
MATERIAL & SIZE 2024-T4 ALUMINUM		APPR NL11	5/17/62	TITLE BLOCK, FACE PLATE SUPPORT	
CHECK				SCALE 1/1	REV LTR
DRAWN	Asubin	SIGNATURE	DATE	NUMBER A-522	
FINISH	Asm				

REVISIONS		
REV	DESCRIPTION	DATE



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES ± # ± .005 ± #	APPLICATION NEXT ASSEMBLY USED ON D-5101	HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS
MATERIAL & SIZE TEFLON	TITLE BEARING, TEFLON, FLANGED.	
FINISH NONE	SCALE 4/1	
APPR NLH	NUMBER A-5213	
CHECK	REV LTR	
DRAWN W. J. Walker	DATE 2-28-62	
SIGNATURE		

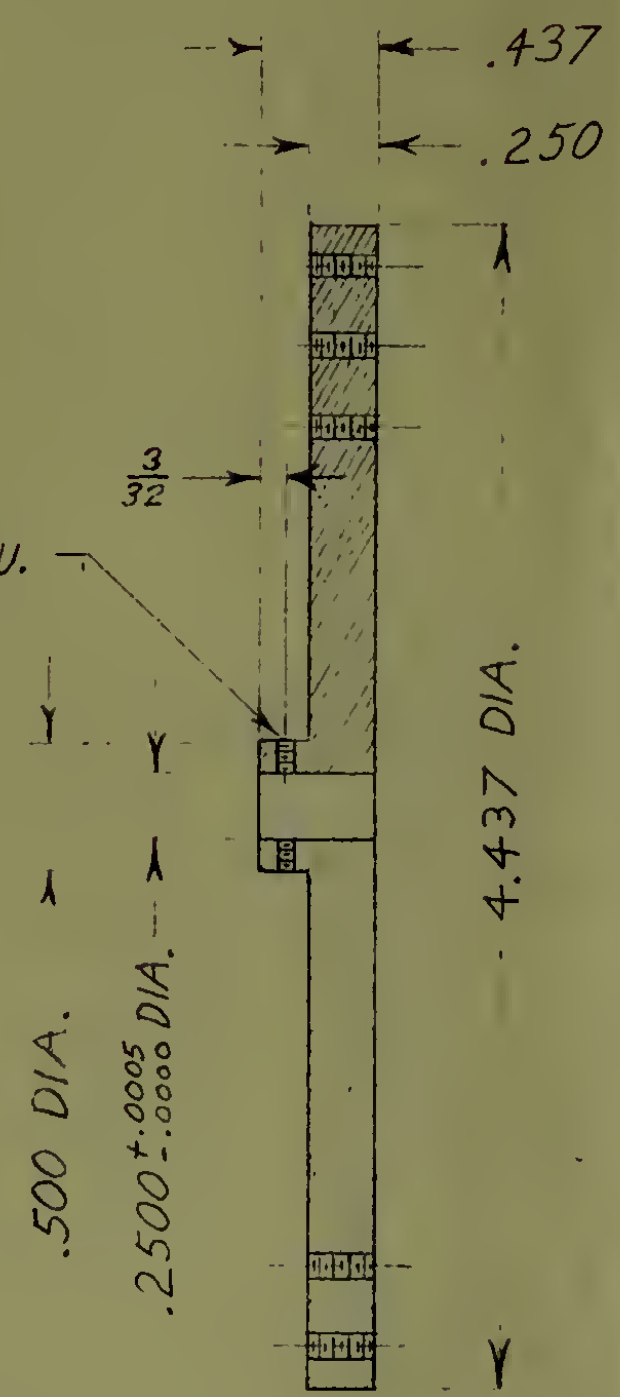
REV
NUMBER
B-5214

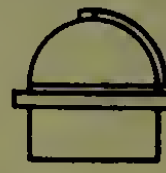
REVISIONS			
REV	DESCRIPTION	DATE	APPROVED



4-40NC-2, THRU.
65 HOLES REQ'D. AS SHOWN.
26 HOLES ON 2.062 D.B.C.
26 HOLES ON 1.750 D.B.C.
13 HOLES ON 1.437 D.B.C.

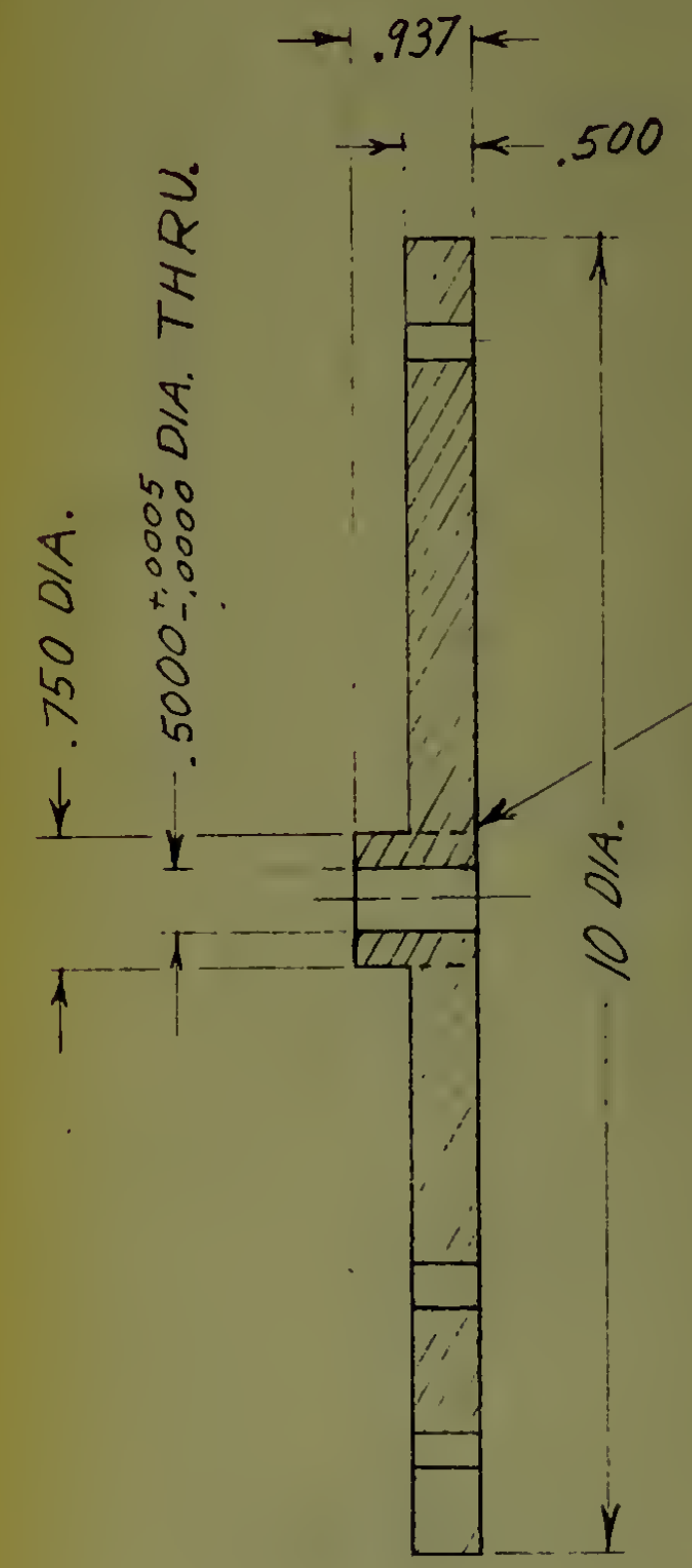
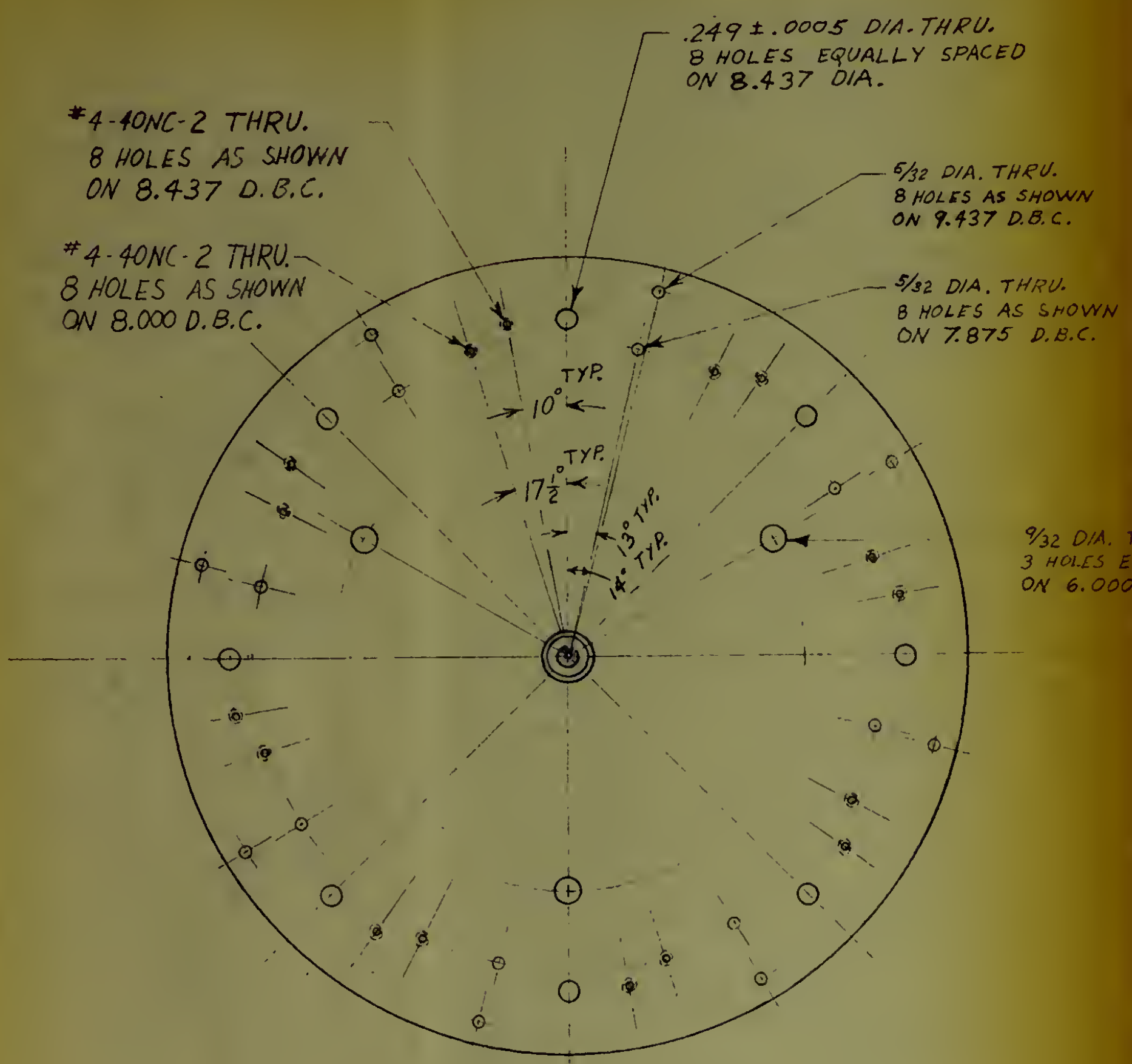
* 4-40NC-2, THRU.
2 HOLES REQ'D.



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON		NO	QUAN	NUMBER	DESCRIPTION	
FRACTIONS	DECIMALS	ANGLES	APPLICATION		 HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
± 1/64	± .005	± 15'	NEXT ASSEMBLY			
MATERIAL & SIZE			D-5101		USED ON	
2024-T4 ALUMINUM			APPR	NLH	5/17/62	TITLE WHEEL, MAGNET
FINISH CLEAR ANODIZED			CHECK			
			DRAWN	Mel Wallace	2-23-62	SCALE 1/1
			SIGNATURE		DATE	
					NUMBER B-5214	REV LTR

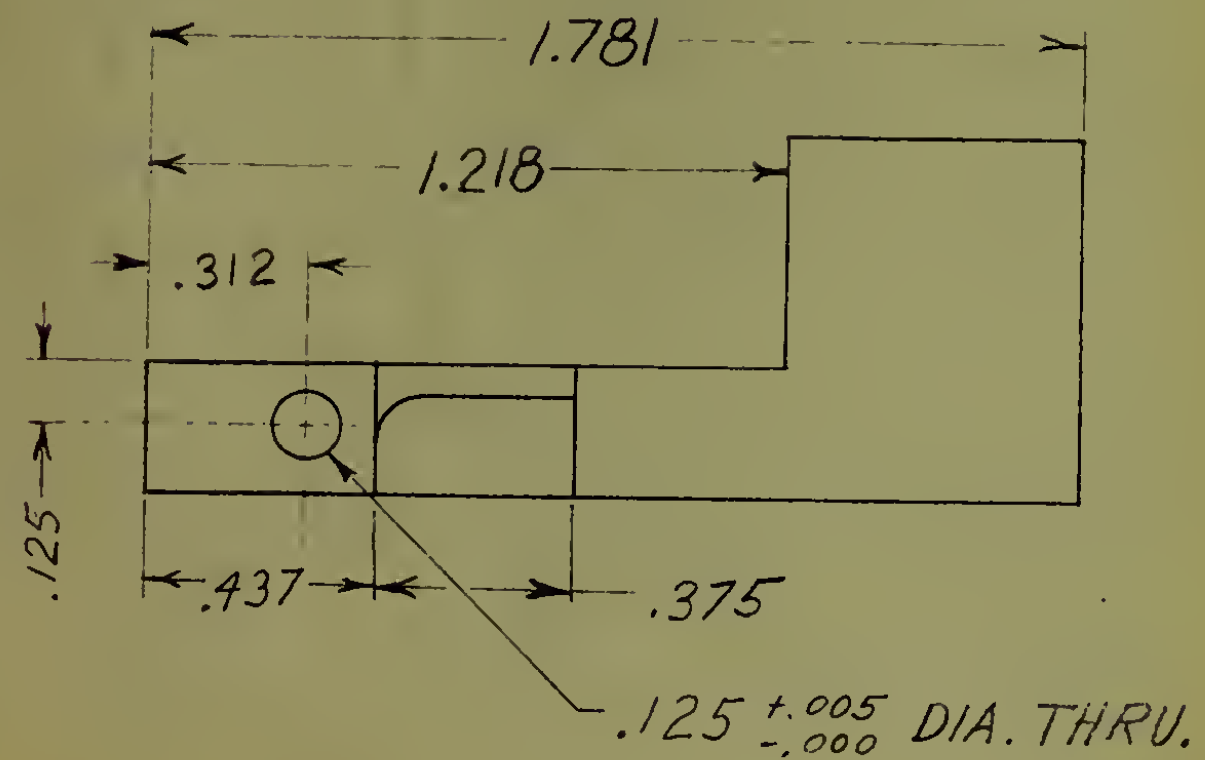
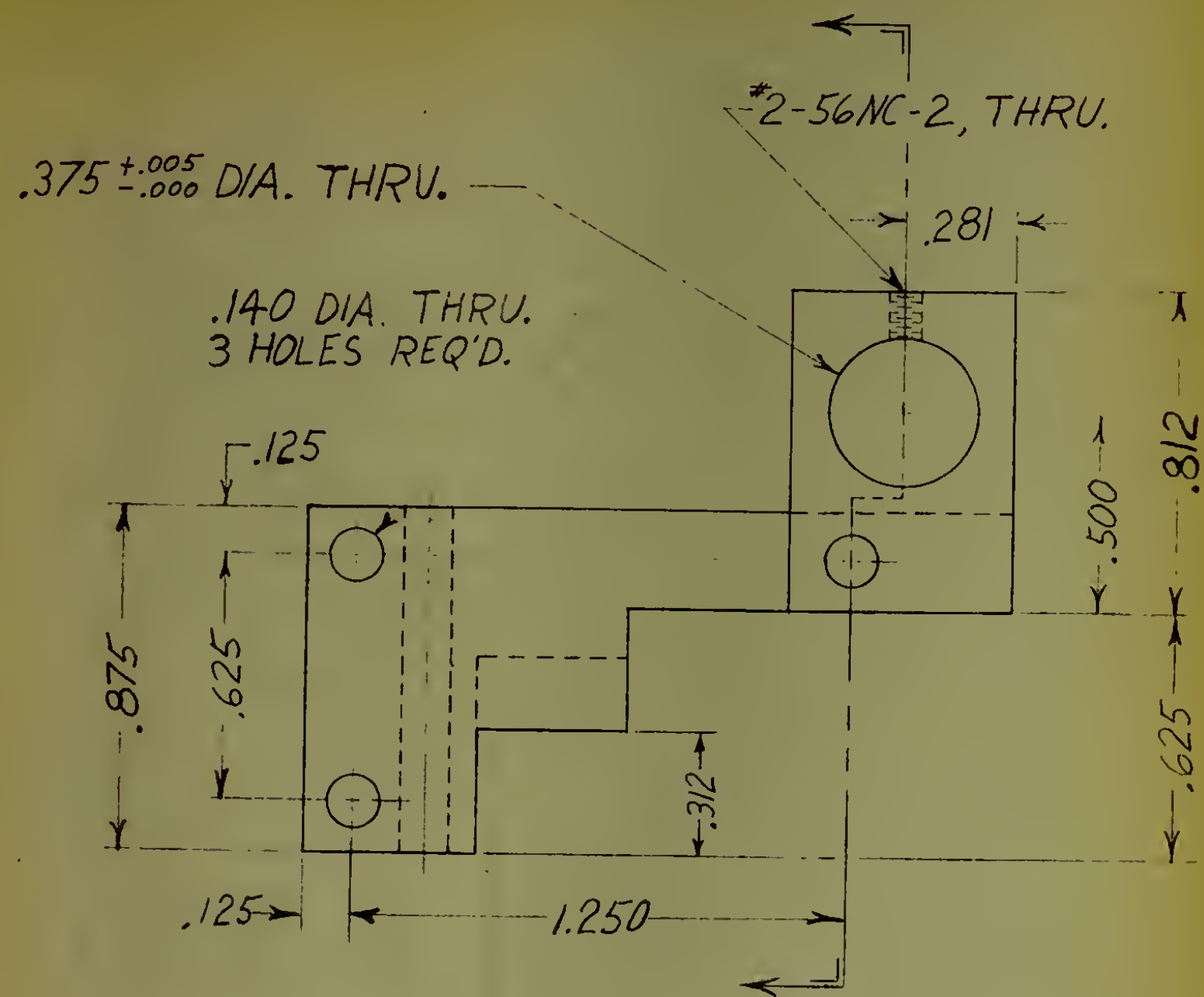
REV
NUMBER
B-5215

REVISIONS		DATE	APPROVED
REV	DESCRIPTION		

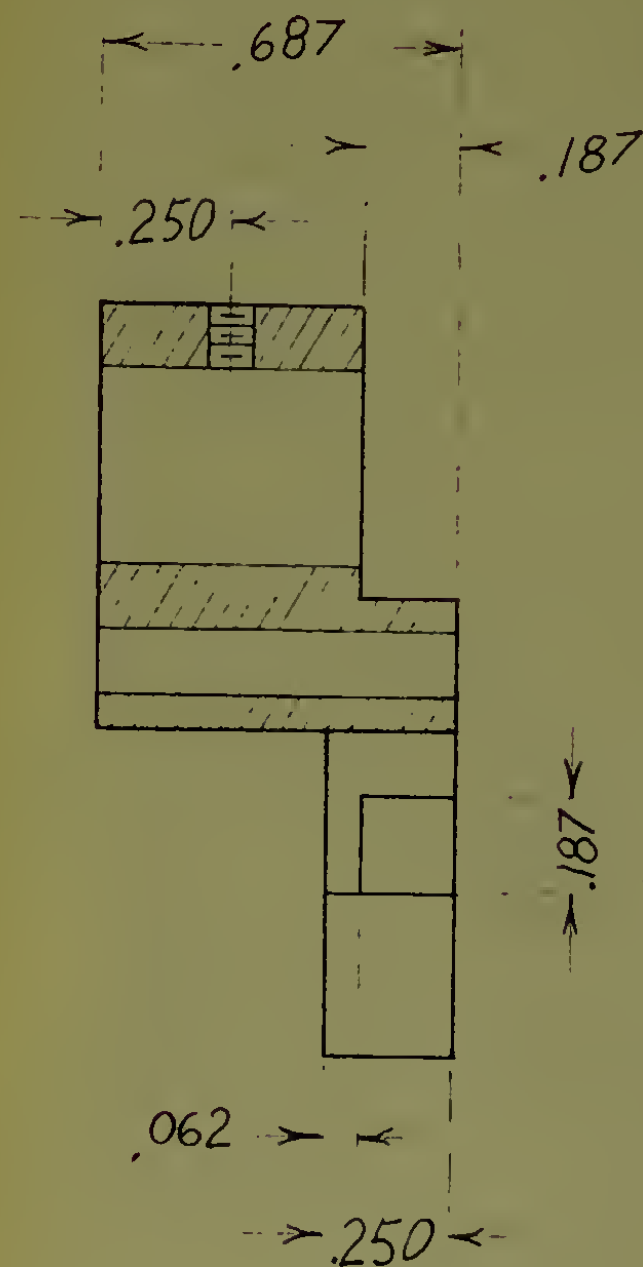


NOTE:
TWO PIECES MAY BE USED IF
FINAL BORING IS DONE AFTER
ASSEMBLY.

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES ± 1/64 ± .025 ± 15'		NO		QUAN	NUMBER	DESCRIPTION	
		APPLICATION					
MATERIAL & SIZE 2024-T4 ALUMINUM		NEXT ASSEMBLY		USED ON		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
		D-5101					
FINISH BLACK ANODIZE		APPR	NLH	5/17/62		TITLE PLATE, BASE.	
		CHECK					
		DRAWN	McWalker	3-7-62		SCALE	NUMBER
		SIGNATURE		DATE		1/2	B-5215
						REV	LTR



REV
NUMBER
B-5216



UNLESS OTHERWISE SPECIFIED
DIMENSIONS IN INCHES - TOLERANCES ON
FRACTIONS DECIMALS ANGLES
± # ± .005 ± #

MATERIAL & SIZE

2024-T4 ALUMINUM

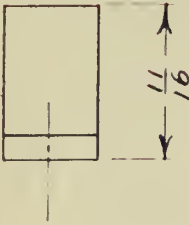
FINISH

BLACK ANODIZE

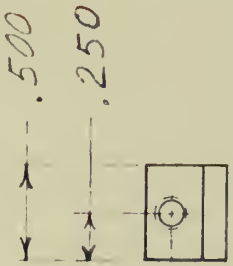
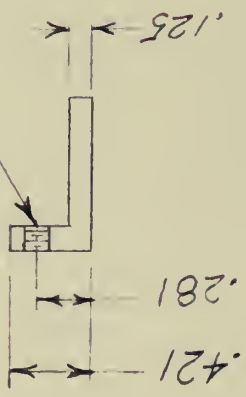
NO	QUAN	NUMBER
APPLICATION		
NEXT ASSEMBLY		USED ON
D-5101		
APPR	NLH	5/17/62
CHECK		
ORAWN	Wolfe	2-5-62
SIGNATURE		DATE

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED

HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
TITLE BRACKET, TRANSDUCER	
SCALE 2/1	NUMBER B-5216
REV LTR	



#6-40NF-2 THRU.



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES	
$\pm \frac{1}{64}$	$\pm .005$
MATERIAL & SIZE	
2024-T4 ALUMINUM	
FINISH	
NONE	

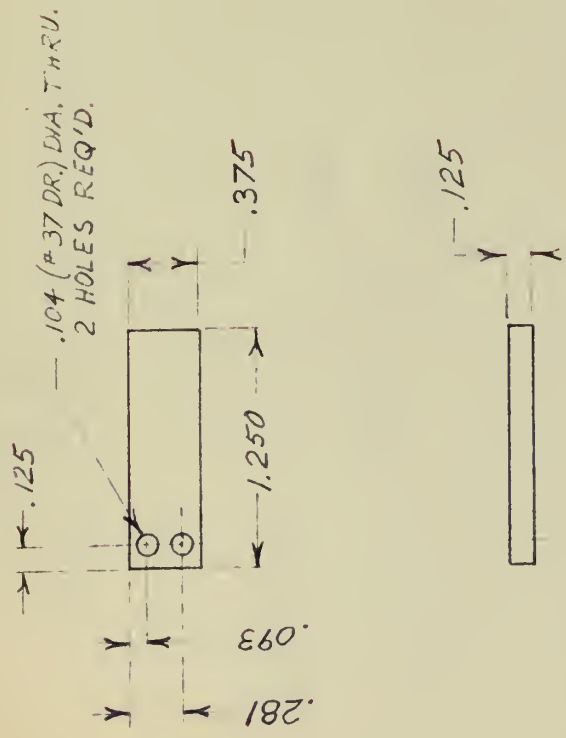
APPLICATION	
NEXT ASSEMBLY	USED ON
APPR	
CHECK	
DRAWN	
DATE	
SIGNATURE	
DATE	

HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
TITLE	
STOP, TEST STATION	
SCALE	1/1
NUMBER	A-5223
REV LTR	

REVISIONS	
REV	DESCRIPTION
DATE	
APPROVED	

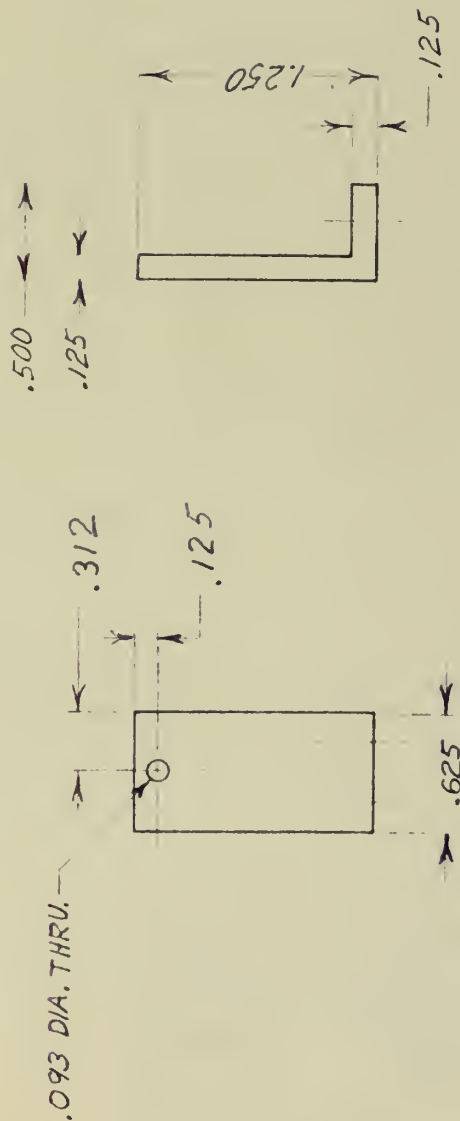
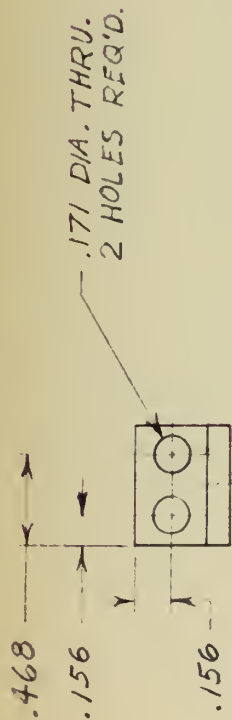
REVISIONS

REV	DESCRIPTION	DATE	APPROVED

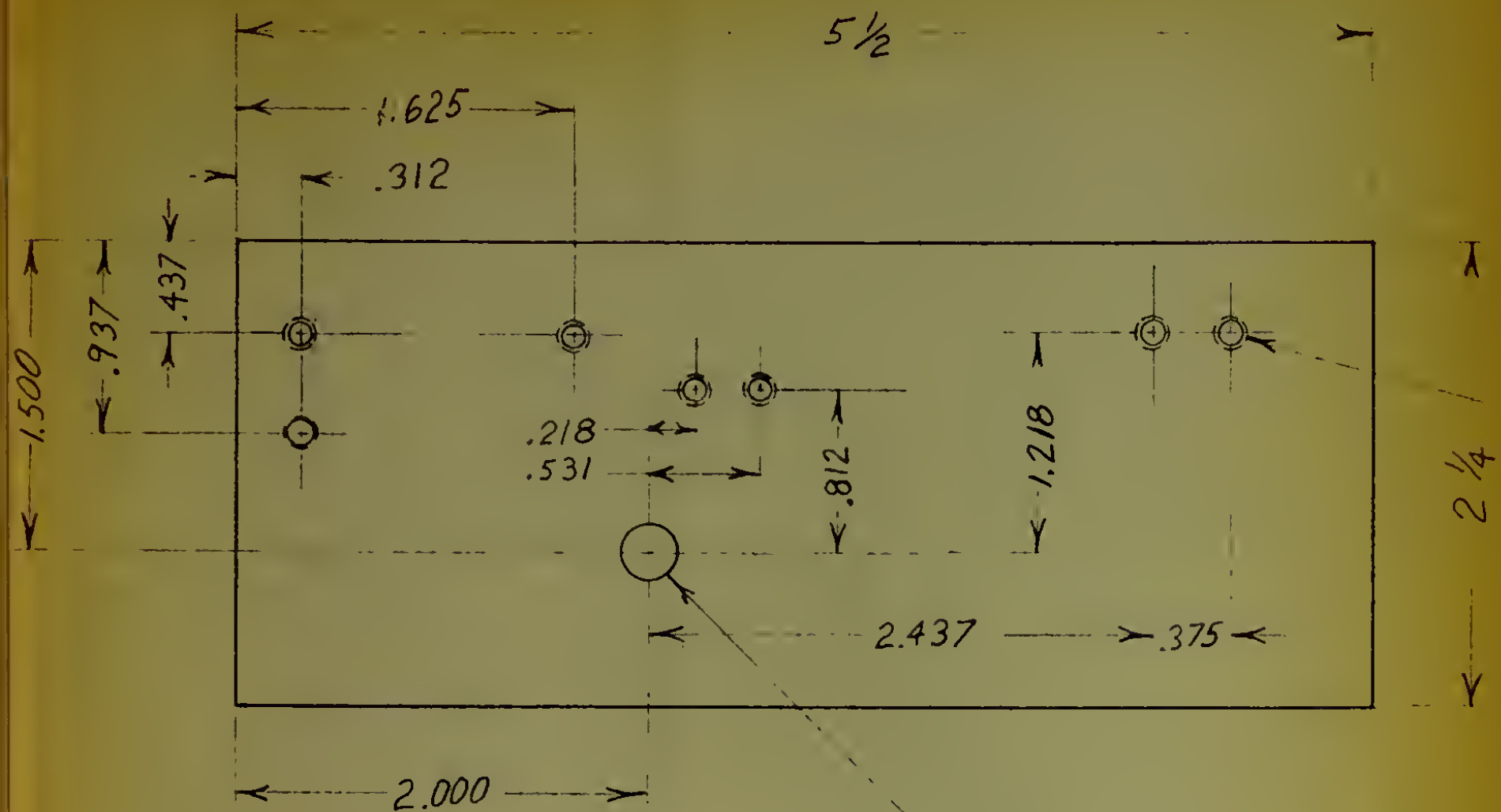


UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm \frac{1}{64}$ $\pm .005$ $\pm \frac{1}{4}$		APPLICATION NEXT ASSEMBLY USED ON B-5105		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
MATERIAL & SIZE 303 STAINLESS STEEL		APPR NLH	CHECK	TITLE RETAINER, TEST STATION	
FINISH VAPOR BLAST		DRAWN H. J. HARRIS	SIGNATURE	SCALE 1/1	REV LTR A-5217

REV	DESCRIPTION	DATE	APPROVED



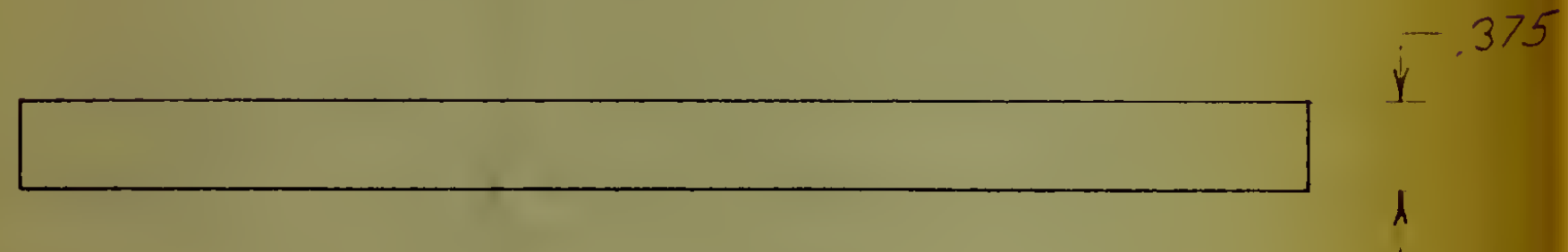
		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS	
TITLE BRACKET, ADJUSTMENT		SCALE 1/1	
NUMBER A-5219		REV LTR 49	
APPLICATION NEXT ASSEMBLY B-5105		USED ON 5/17/62	
APPR NLH	CHECK [Signature]	DATE 3-10-62	
UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm \frac{1}{64}$ $\pm .005$ $\pm \frac{1}{2}$		MATERIAL & SIZE 303 STAINLESS STEEL	
FINISH VAPOR BLAST		SIGNATURE W. W. [Signature]	



REV
NUMBER
B-5220

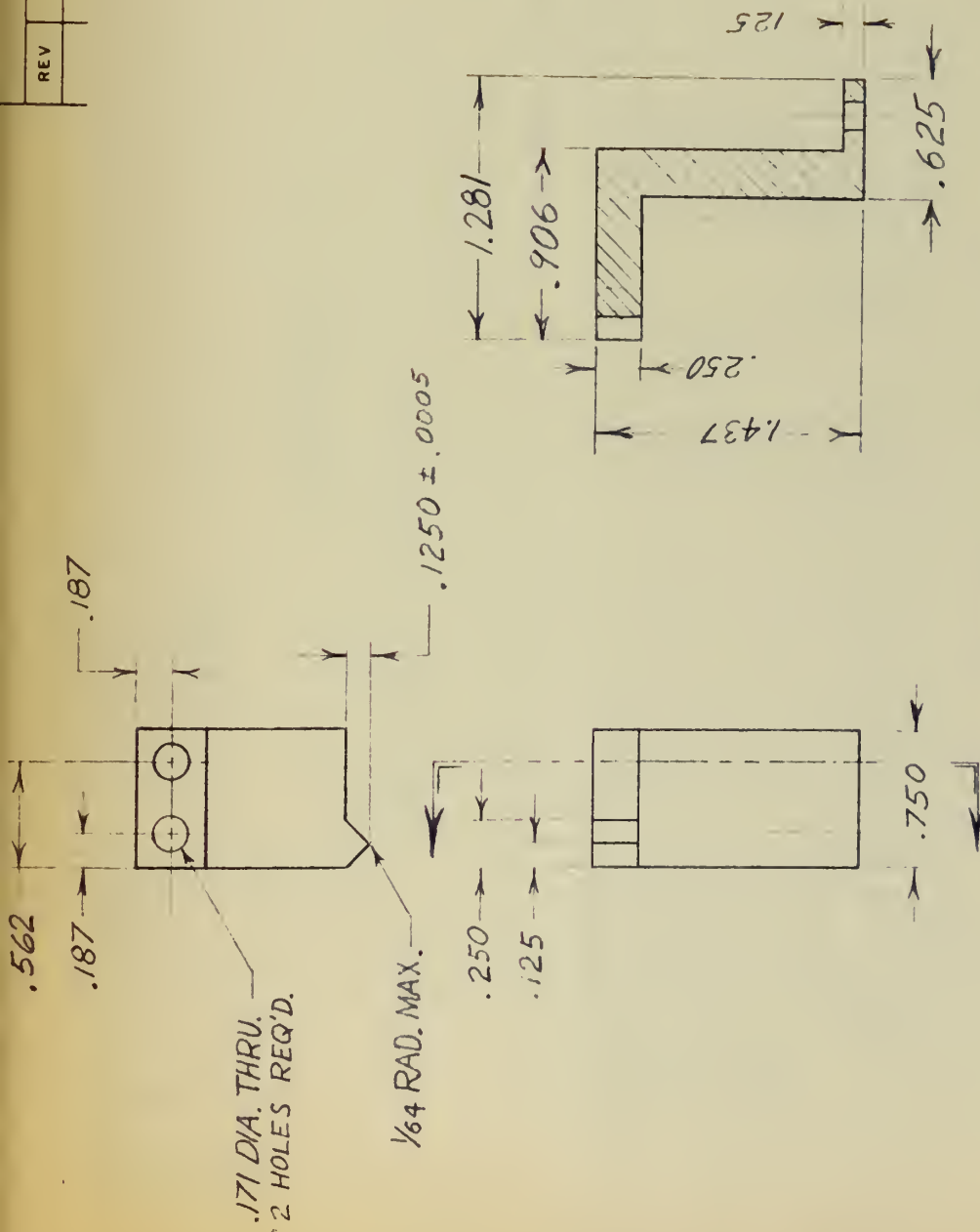
6-32NC-2, THRU.
7 HOLES REQ'D AS
SHOWN.

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED



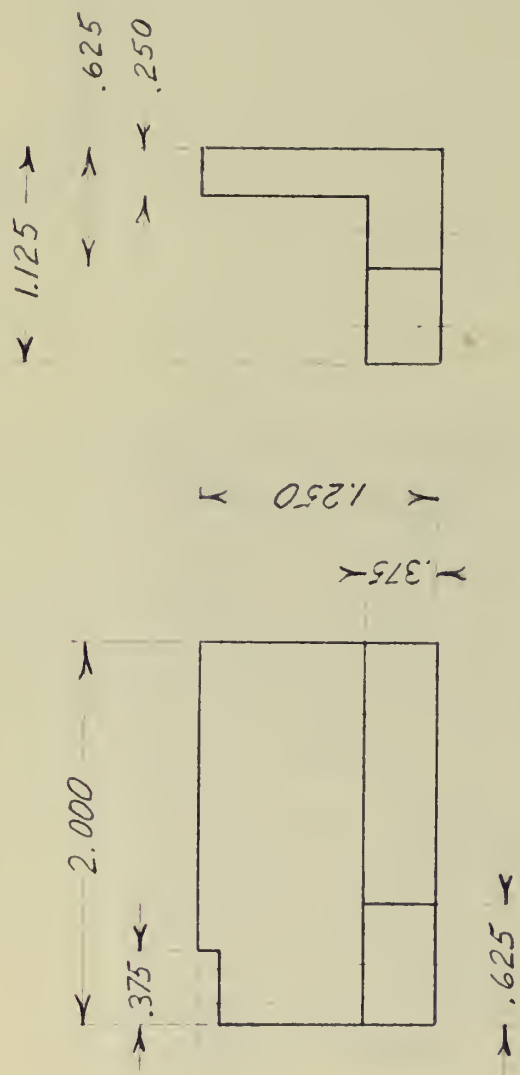
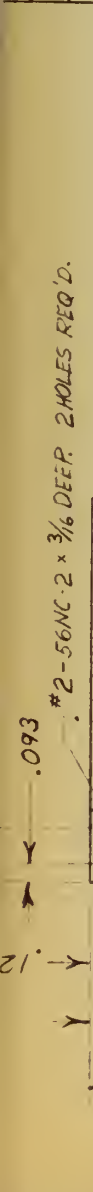
UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON		NO	QUAN	NUMBER	DESCRIPTION	
FRACTIONS	DECIMALS	ANGLES	APPLICATION			HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS
± 1/64	± .005	± #	NEXT ASSEMBLY	USED ON		
MATERIAL & SIZE			B-5105		TITLE	
2024-T4 ALUMINUM			APPR	NLH	5/17/62	BASE
FINISH			CHECK			
BLACK ANODIZED			DRAWN	Mc Walker	3-10-62	SCALE
			SIGNATURE		DATE	NUMBER
						B-5220
					1/1	REV LTR

REVISIONS			REV	DESCRIPTION	DATE	APPROVED

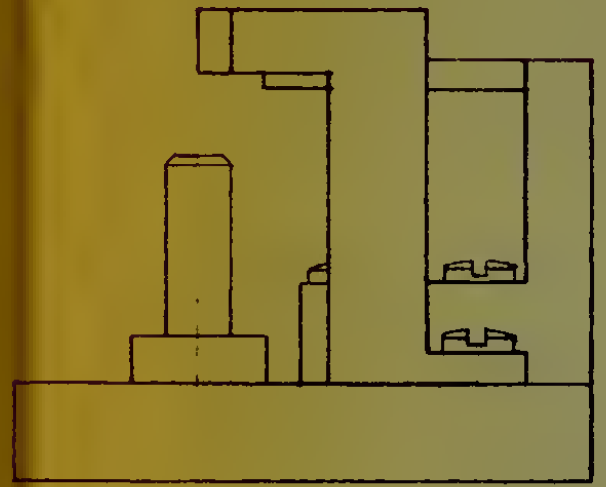
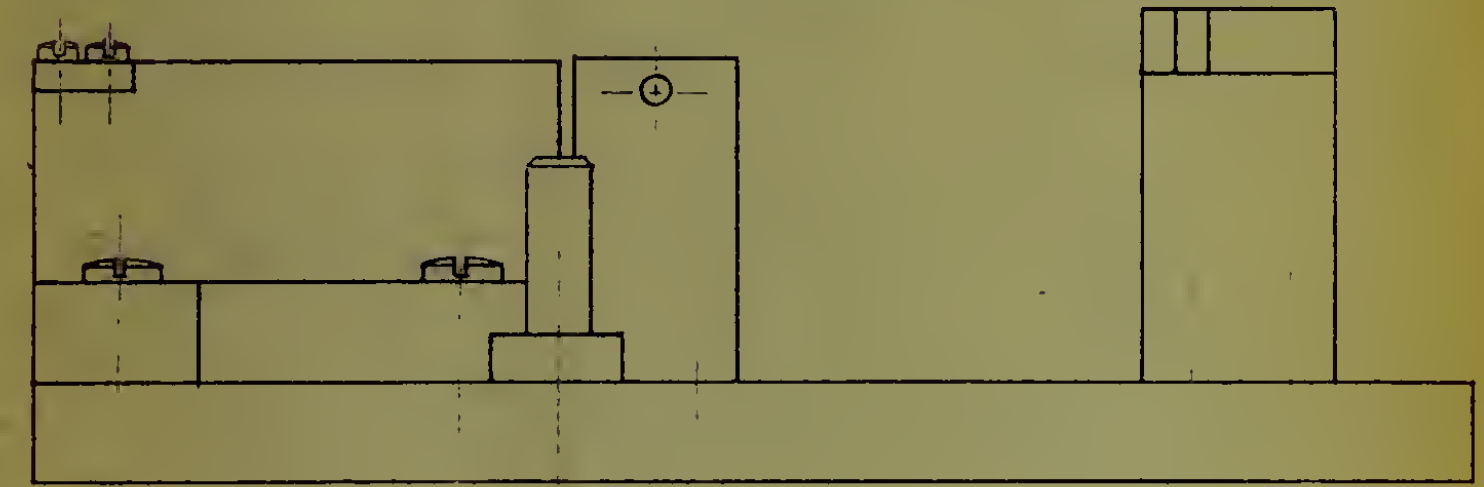
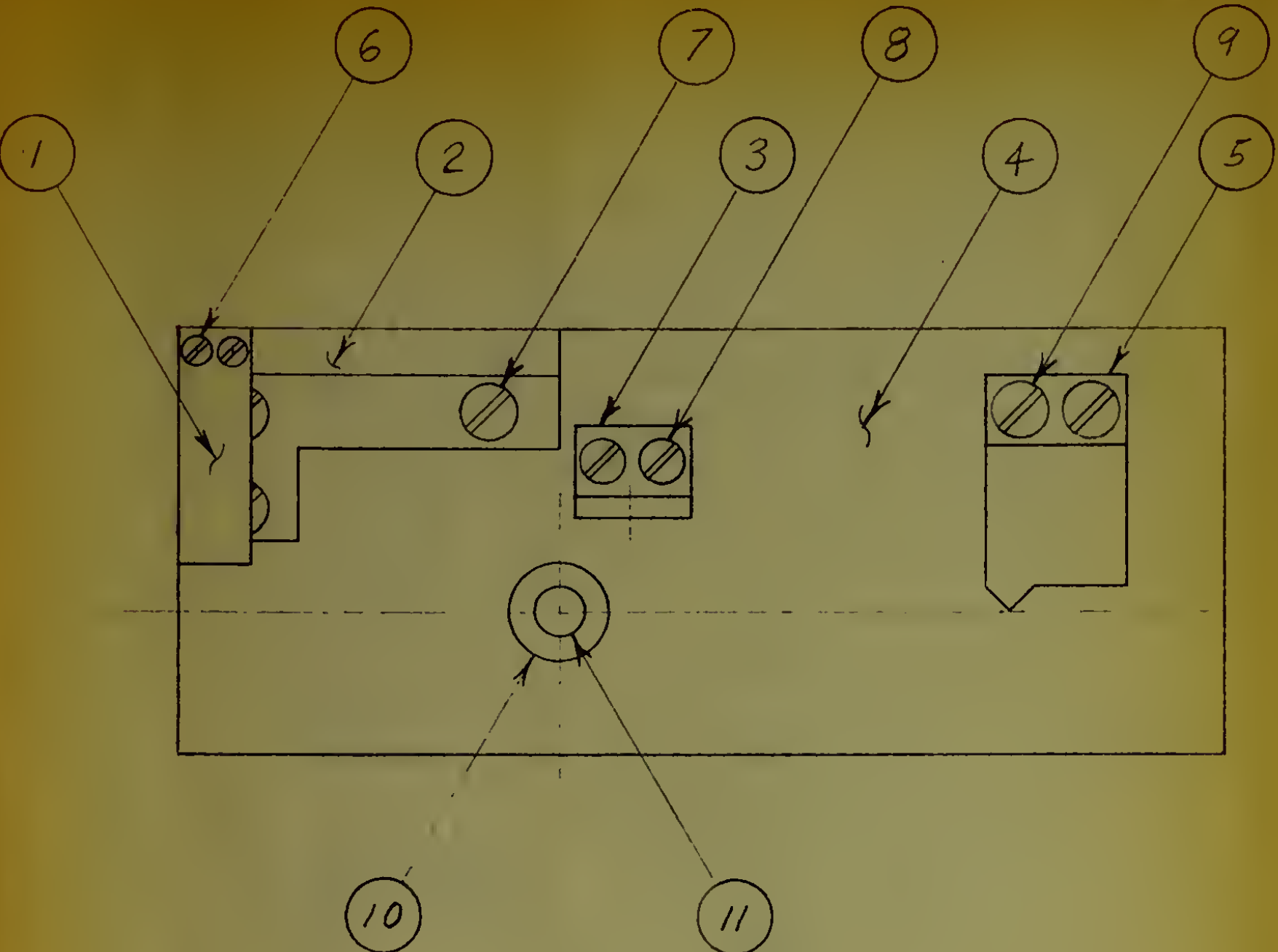


 HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS		TITLE	
		KNIFE EDGE.	
APPLICATION NEXT ASSEMBLY USED ON B-5105		SCALE	REV LTR
APPR NLH 5/17/62 CHECK DRAWN <i>[Signature]</i> 3-10-62 SIGNATURE DATE		1/1	A-5221
UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm$ $\frac{1}{64}$ $\pm$.005 $\pm$ $^{\circ}$ MATERIAL & SIZE 303 STAINLESS STEEL		FINISH VAPOR BLAST	

REV	DESCRIPTION	DATE	APPROVED



UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON FRACTIONS DECIMALS ANGLES $\pm \frac{1}{64}$ $\pm .005$ $\pm \frac{1}{4}$		APPLICATION NEXT ASSEMBLY B-5105		USED ON 5/17/62	
MATERIAL & SIZE 303 STAINLESS STEEL		APPR N/LH		5/17/62	
FINISH VAPOR BLAST		CHECK N/LH		3-10-62	
DRAWN N/LH		SIGNATURE N/LH		DATE 3-10-62	
SCALE 1/1		NUMBER A-5222		REV LTR	
TITLE SUPPORT, SIDE.		HARVARD COLLEGE OBSERVATORY CAMBRIDGE 38 MASSACHUSETTS			



REV
NUMBER
B-5105

REVISIONS			
REV	DESCRIPTION	DATE	APPROVED

11	1		SHAFT, PIC # A6-13.
10	1		SPACER, PIC # B8-12.
9	2		SCREW, #6-32NC-2 x 3/8 LG, BIND.HD., S.S.
8	2		SCREW, #6-32NC-2 x 3/8 LG, FIL. HD., S.S.
7	3		SCREW, #6-32NC-2 x 5/8 LG, BIND.HD., S.S.
6	2		SCREW, #2-56NC-2 x 1/4 LG, FIL. HD. S.S.
5	1	A-5221	KNIFE EDGE.
4	1	B-5220	BASE.
3	1	A-5219	BRACKET, ADJUSTMENT.
2	1	A-5222	SUPPORT, SIDE.
1	1	A-5217	RETAINER, TEST STATION.
NO	QUAN	NUMBER	DESCRIPTION

UNLESS OTHERWISE SPECIFIED
DIMENSIONS IN INCHES - TOLERANCES ON
FRACTIONS DECIMALS ANGLES
± .001 ± .001 ± .001

MATERIAL & SIZE
H

FINISH
H

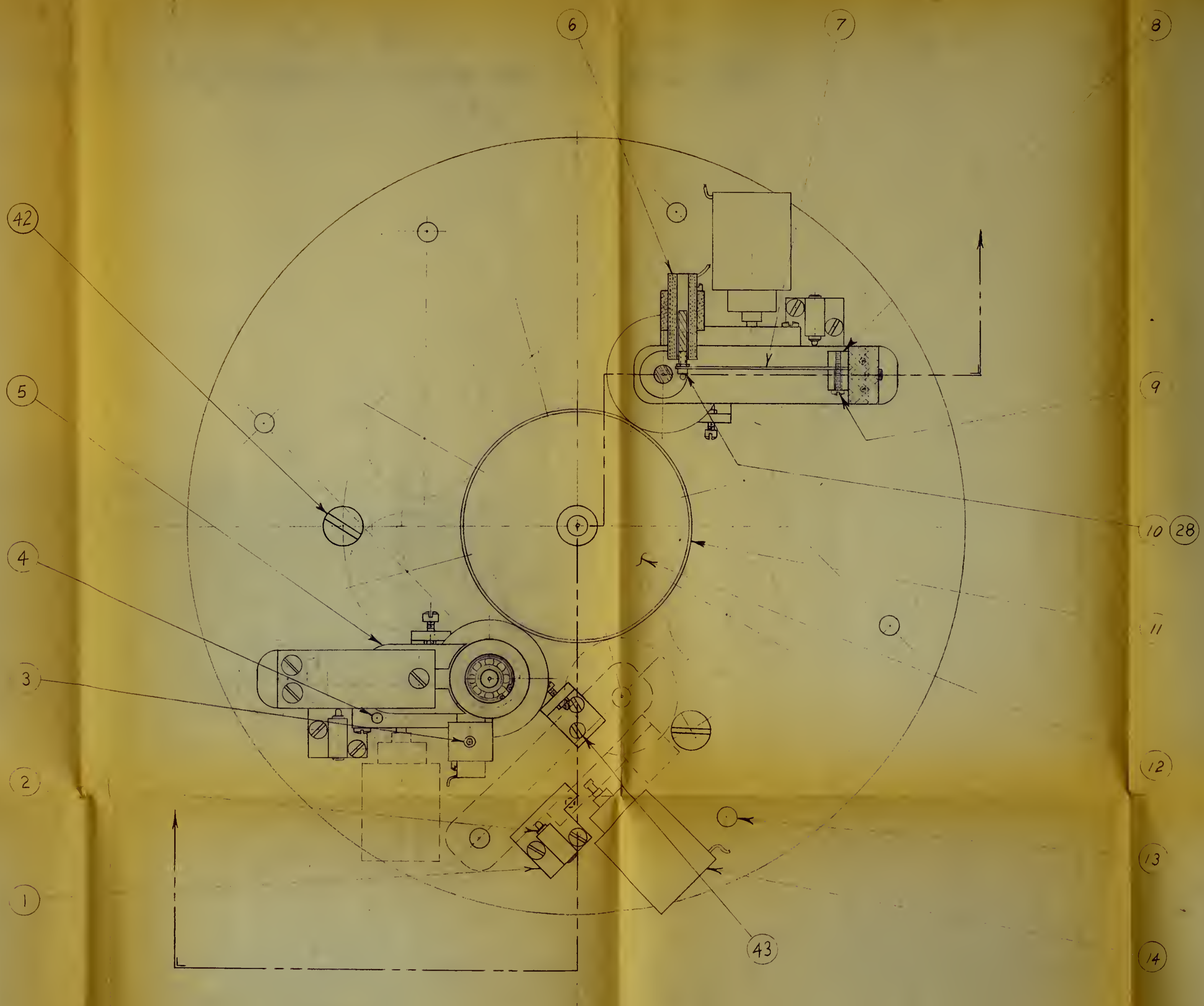
APPLICATION	
NEXT ASSEMBLY	USED ON
APPR	NLH 5/17/62
CHECK	
DRAWN	Mel Walker 3-9-62
SIGNATURE	DATE

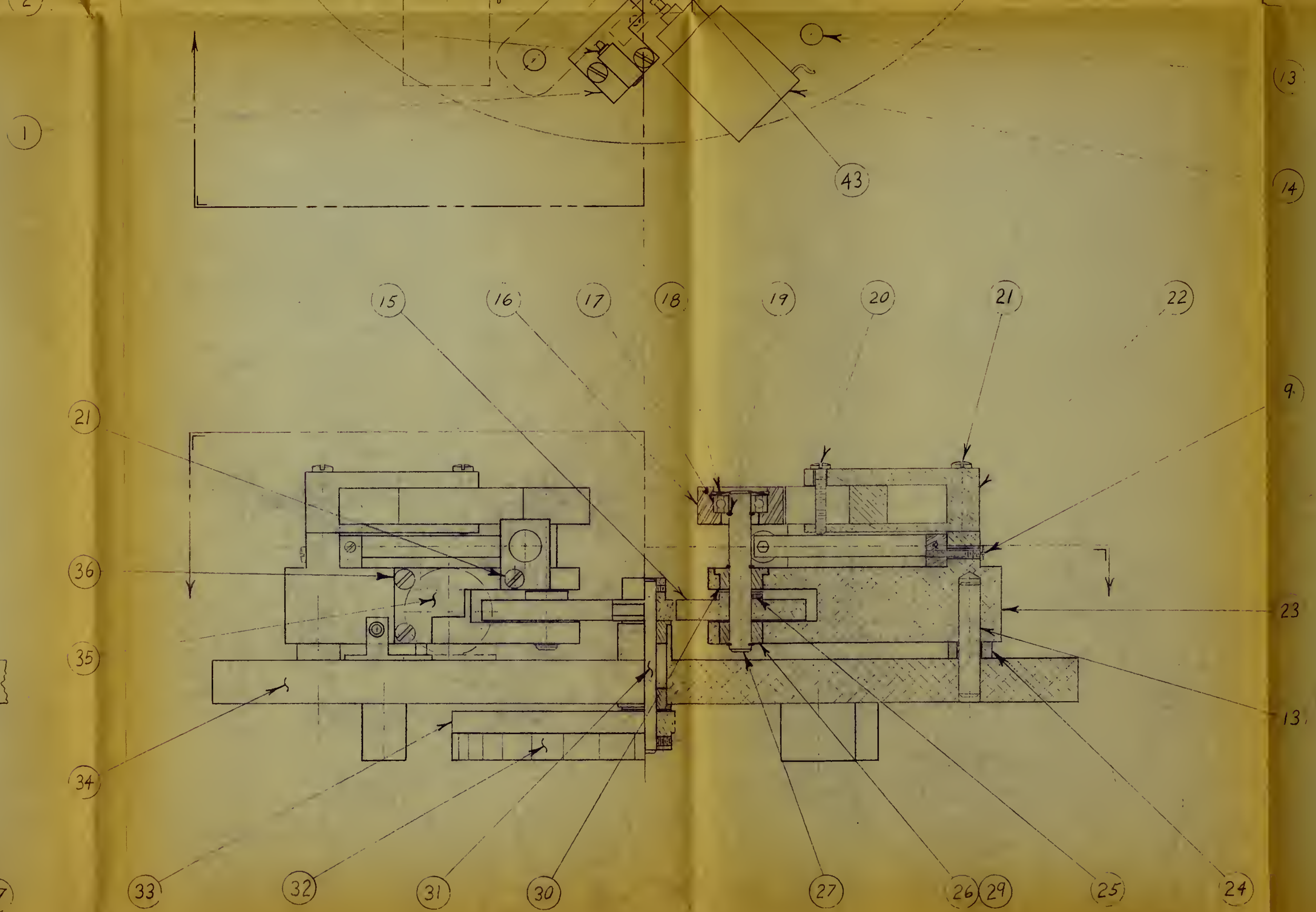
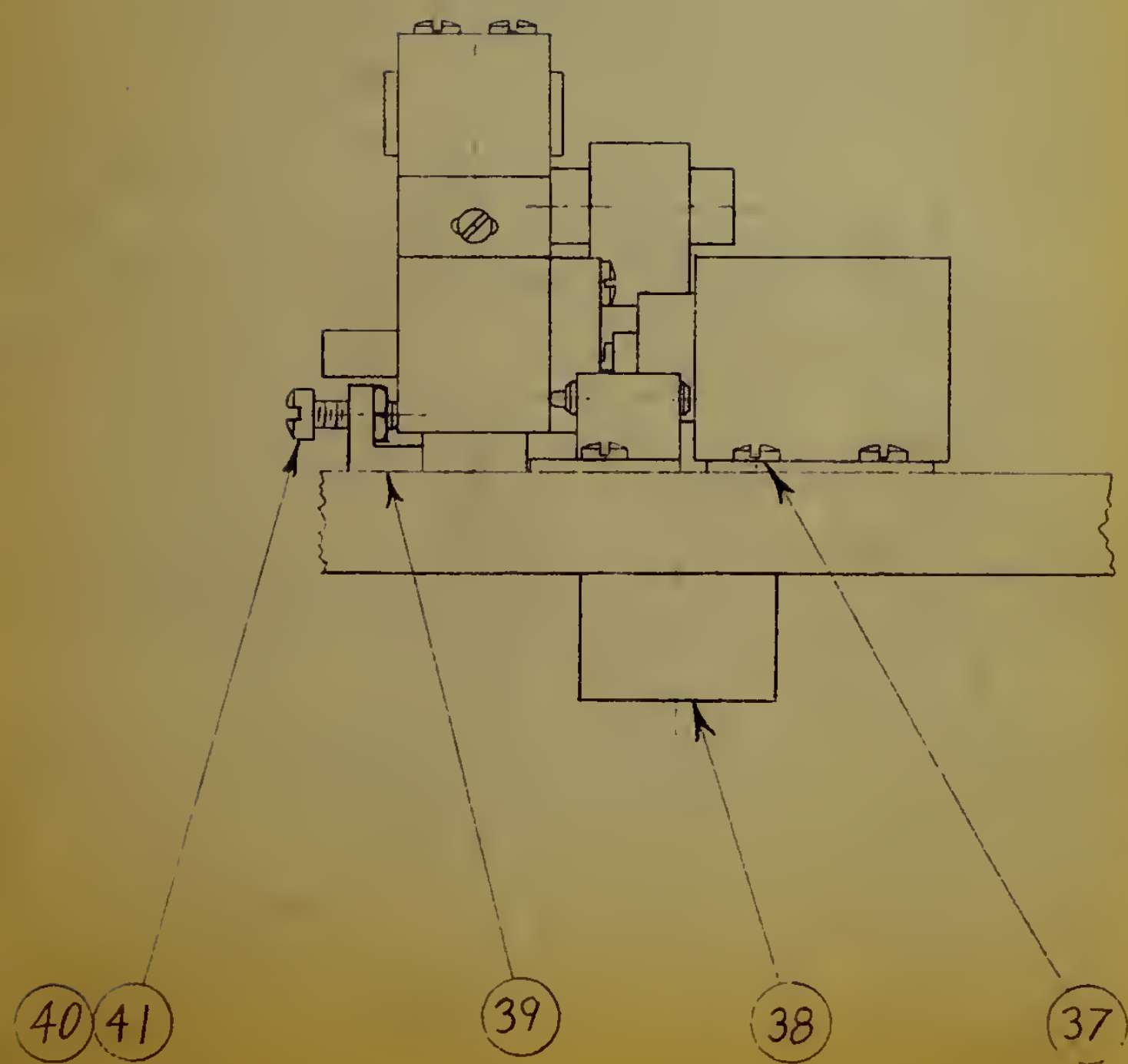


**HARVARD COLLEGE
OBSERVATORY**
CAMBRIDGE 38 MASSACHUSETTS

TITLE
JIG, TEST STATION CALIBRATION

SCALE 1/1	NUMBER B-5105	REV LTR
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REVISIONS			
REV		DESCRIPTION	DATE
43	16		SCREW, 2-56 NC-2 x 3/8 LG, FLAT HD., SS.
42	3		SCREW, 1/4-20 NC-2 x 3/4 LG., BIND. HD., S.S.
41	8		NUT, HEX, 6-40 NF-2, S.S.
40	8		SCREW, 6-40 NF-2 x 3/8 LG, FIL. HD., S.S.
39	8	A-5223	STOP, TEST STATION.
38	3	A-5212	BLOCK, BASE PLATE SUPPORT.
37	32		SCREW, #4-40 NC-2 x 1/4 LG, BIND. HD., SS.
36	16		SCREW, #4-40 NC-2 x 1/2 LG, BIND. HD., S.S.
35	8	B-5216	BRACKET, TRANSDUCER.
34	1	B-5215	PLATE, BASE.
33	1	B-5214	WHEEL, MAGNET, DRIVEN.
32	130		MAGNET, ARNOLD ENG. # 3241.
31	1		SHAFT, DRIVE, PIC # A6-20.
30	10	A-5213	BEARING, TEFLON, FLANGED.
29	16		WASHER, THRUST.
28	8		NUT, HEX, # 1-72 NF-2, SS.
27	8	A-5211	SHAFT, TEST BEARING.
26	8	A-5210	BEARING, TEFLON, PLAIN.
25	10		SET SCREW, #2-56 x 1/8 LG, SK. HD., NO-MAR
24	8		SPACER, PIC # B8-12
13	8	B-5209	BLOCK, BEARING TEST STATION.
12	8	B-5208	BRACKET, MAGNET.

43	16		SCREW, 2-56NC-2 x 3/8 LG, FLAT HD., SS.
42	3		SCREW, 1/4-20NC-2 x 3/4 LG., BIND. HD., S.S.
41	8		NUT, HEX, 6-40NF-2, S.S.
40	8		SCREW, 6-40NF-2 x 3/8 LG, FIL. HD., S.S.
39	8	A-5223	STOP, TEST STATION.
38	3	A-5212	BLOCK, BASE PLATE SUPPORT.
37	32		SCREW, #4-40NC-2 x 1/4 LG., BIND. HD., SS.
36	16		SCREW, #4-40NC-2 x 1/2 LG, BIND. HD., S.S.
35	8	B-5216	BRACKET, TRANSDUCER.
34	1	B-5215	PLATE, BASE.
33	1	B-5214	WHEEL, MAGNET, DRIVEN.
32	130		MAGNET, ARNOLD ENG. # 3241.
31	1		SHAFT, DRIVE, PIC # A6-20.
30	10	A-5213	BEARING, TEFLON, FLANGED.
29	16		WASHER, THRUST.
28	8		NUT, HEX, # 1-72NF-2, SS.
27	8	A-5211	SHAFT, TEST BEARING.
26	8	A-5210	BEARING, TEFLON, PLAIN.
25	10		SET SCREW, #2-56 x 1/8 LG, SK. HD., NO-MAR
24	8		SPACER, PIC # B8-12
23	8	B-5209	BLOCK, BEARING TEST STATION.
22	8	B-5208	BRACKET, MAGNET.

NUMBER	REV
D-5101	

21	24		SCREW, #4-40NC-2 x 7/8 LG., FIL. HD., SS.
20	8		SCREW, #4-48NF-2 x 3/4 LG., FIL. HD., S.S.
19	24		SNAP RING, TRUARC # 5103-25.
18	8		SNAP RING, TRUARC # N5001-62.
17	8		TEST BEARING.
16	8	B-5207	HOUSING, TEST BEARING.
15	8	A-5206	WHEEL, DRIVEN.
14	8		SOLENOID, GUARDIAN 22 INT. 24DC.
13	8		PIN, DOWEL, PIC # A6-13.
12	1	B-5205	WHEEL, DRIVING.
11	1		O-RING, PARKER # 2-334, BUNA-N.
10	8	A-5204	KNIFE EDGE.
9	16		SCREW, #1-72NF-2 x 1/2 LG., BIND. HD., SS.
8	8	A-5203	BLOCK, SPRING.
7	8	A-5202	SPRING.
6	8		TRANSDUCER, LINEAR SYN # 595DT-050.
5	8		MAGNET, LOAD.
4	8		PIN, STOP, PIC # A1-13.
3	8		SET SCREW, NO-MAR # 2-56 x 5/32 LG.
2	8		PLUNGER, VLIER # S51.
1	8	B-5201	BRACKET, PLUNGER.

NO	QUAN	NUMBER	DESCRIPTION
APPLICATION			
NEXT ASSEMBLY		USED ON	
APPR	NLH	5/17/62	
CHECK			
DRAWN	McWalker	11/21/61	
SIGNATURE		DATE	



HARVARD COLLEGE
OBSERVATORY
CAMBRIDGE 38 MASSACHUSETTS

TITLE	TEST APPARATUS, BEARING, HIGH VACUUM.	SCALE	NUMBER	REV LTR
		1/1	D-5101	

UNLESS OTHERWISE SPECIFIED DIMENSIONS IN INCHES - TOLERANCES ON		
FRACTIONS	DECIMALS	ANGLES
± #	± #	± #
MATERIAL & SIZE		
#		
FINISH		
#		

thesW22214

The design and construction of a high va



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